S·A·E JOURNAL

PUBLISHED BY THE SOCIETY OF AUTOMOTIVE ENGINEERS, INC.

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Vol. 38 — JANUARY, 1936 — No. 1

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Publication Office, 56th and Chestnut Sts., Philadelphia, Pa.; Editorial and Advertising Departments at the headquarters of the Society, 29 West 39th St., New York, N. Y. Western Advertising Office, Room 2-136 General Motors Bldg., Detroit, Mich.

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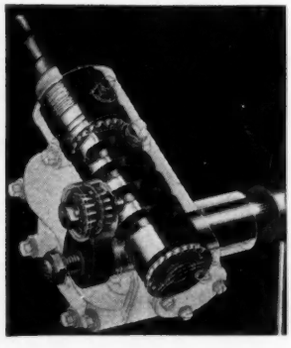

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A Study of the Economical Loading and Operation of Motor Trucks

By A. J. Scaife
Consulting Field Engineer, The White Motor Co.

MANY organizations and individuals have been trying to get a simple formula for measuring the economy of motor trucking from a load-carrying standpoint; that is to say, is it more economical to buy a low-priced truck, load it to the limit for a few thousand miles, discard it and buy a new one, instead of buying a higher-priced truck, and run it three or four times farther, even up to the point of obsolescence, before buying a new one?

To make an intelligent analysis, it is well to consider first the type of truck that should be used to do the work required in the most efficient manner and then compare it with a vehicle that will do the work after a fashion. In making a study of this subject, we often find that all the factors have not been taken into consideration.

When actual operating records are compared in the same organization, doing the same kind of work, the results are most interesting. And it is easy to see that it is poor economy to buy a motor truck and operate it out of its class by overloading, not only from an economic standpoint, but also from a safety and highway-hazard standpoint. Failure on the highway, loss of time, delay in delivery, and the like, are all factors that should be considered.

THE motor-truck industry has been confronted with the problem of economical loading of motor trucks. This subject has also been of similar interest to the users of motor vehicles, as well as to legislatures and highway officials. A common method of measuring the carrying ability, together with the economic operation of a motor truck, has been a subject under discussion by the Society for a long period. The method of rating the carrying capacity of a commercial vehicle is more or less guess-work, and making comparisons of the capacities of the different motor trucks by the published rating is only confusing. There has been no real method by which the true rating of any commercial vehicle can be established.

In making an analysis of the July, 1935, issue of the *Commercial Car Journal* we find that the gross rating, when compared with the tonnage rating, is not comparable in any sense of the word, as indicated by the following tabulation:

Tonnage and Gross Rating on a Four-Wheel Truck

Tons	*Gross Rating, Lb.* *From*	*To*
1	7,100	11,000
1½	6,000	12,500
2	11,400	19,000
2½	13,000	26,000
3	16,300	32,000
3½	19,000	25,000
4	19,000	38,000
5	23,000	37,000

One can see by the foregoing that it is practically impossible to make comparisons as to the carrying capacity of vehicles that will be of any value.

The Transportation and Maintenance Activity of the Society has had this subject before the Rating Committee for some time, and it has been trying to get a yardstick that would measure the safe load-carrying capacity of motor trucks of different classifications.

During the summer and fall months of 1934, the Bureau of Public Roads of the United States Department of Agriculture, in cooperation with Johns Hopkins University, the Maryland State Road Commissioner and the Commissioner of Motor Vehicles of Maryland operated two stations on the highway at which the commercial vehicles were weighed and measured. One was located on U. S. Route No. 40 leading north out of Baltimore toward Philadelphia, and the other survey was made on U. S. Route No. 1 running between Baltimore and Washington.

These stations were operated from about the middle of June to the middle of November, and during this period 10,700 vehicles were inspected, of which 7100 were loaded and 3600 were light. The results of this survey are tabulated in the May, 1935, issue of *Public Roads*. This study was

[This paper was presented at the General Transportation and Maintenance Meeting of the Society, Chicago, Oct. 10, 1935.]

Table 1—Frequency Distribution of Gross Weights of All Loaded Single Vehicles, Percentage of Total Observations in Each Capacity Class

Manufacturer's Rated Capacity, Tons	Average Gross Weight, Lb.	Gross Weight, 1000 Lb. 4-6 Per Cent	7-9 Per Cent	10-12 Per Cent	13-15 Per Cent	16-18 Per Cent	19-21 Per Cent	22-24 Per Cent	25-27 Per Cent	28-30 Per Cent	31-33 Per Cent	34-36 Per Cent	37-39 Per Cent	40-42 Per Cent	43-45 Per Cent	Total Observations, Number
1	6,400	70.0	20.0	6.7	.....	3.3	.....	.....	.....	.....	.....	.....	.....	.....	.....	30
1½	10,500	18.4	26.1	22.6	23.7	6.9	1.7	0.4	0.1	0.1	.....	.....	.....	.....	.....	2,920
2	14,400	1.9	14.5	19.3	21.1	23.1	15.4	3.8	0.7	0.2	.....	.....	.....	.....	.....	533
2½	16,300	1.3	7.9	14.6	20.1	26.0	14.6	9.6	4.2	1.1	.....	0.2	0.4	.....	.....	458
3	18,100	0.2	3.3	11.7	15.0	20.4	26.2	14.8	5.6	1.2	0.7	0.2	0.5	0.2	.....	427
3½	20,600	2.3	.....	7.6	15.3	14.5	19.0	17.6	11.4	3.1	1.5	5.3	0.8	0.8	0.8	131
4	21,600	.....	2.9	8.6	8.6	22.8	5.6	8.6	22.8	8.6	8.6	.....	2.9	.....	.....	35
5	21,600	0.4	.....	4.2	6.9	16.8	25.6	25.6	9.5	3.1	1.5	3.0	2.3	1.1	.....	262
Over 5	25,400	2.3	.....	4.7	4.7	11.6	23.2	9.3	9.3	2.3	4.7	4.7	13.9	9.3	.....	43

made in order to ascertain "to what extent do operators exceed the manufacturers' recommended loading." The results of this survey were to be studied in order to analyze intelligently the requirements for road construction as to providing clearance and strength in the design of highways and highway projects; also, in order that highway economists could allocate properly the highway costs of the various classes of vehicles, and so that the manufacturers of both motor vehicles and tires could have a true picture of just what the operators are doing with reference to loading under actual operating conditions.

The information is also for legislatures and highway administrators to be used for restricting motor vehicles as to size and weight and, if possible, to get a more uniform method of applying taxes to various sizes of vehicles in order to sustain a highway program. It was practically impossible for the Department to get the information it desired that could be used authoritatively without setting up a so-called "roadside clinic." The personnel consisted of six men, four of whom were uniformed officers, and the others were used for making tabulations, using a platform scale for weighing the various vehicles. In order to prevent duplications, each vehicle operator was given an identification card so that no vehicle was recorded more than once while loaded or more than once while empty during any two-week period. All commercial vehicles were stopped and, if the operator had an identification card, it was checked as to whether the vehicle had been weighed both loaded and empty. If so, he was allowed to go on. If not, the vehicle was weighed; first, the entire vehicle, and then the different axles. At the same time, the overall length, width, height and wheelbase were determined, and the manufacturer's marker and the manufacturer's rated capacity were noted.

At this point it would be well to state that the laws of Maryland, briefly, are: Width, 96 in.; length, unrestricted; height, unrestricted; gross weights, single unit having four wheels, 25,000 lb.: single unit having six wheels and three axles, 40,000 lb.: combination of two vehicles, tractor and semi-trailer or tractor and full trailer, combined weight, 40,000 lb. In order to get a true picture of operating conditions, no attention was paid to violation of these laws, nor were any penalties of any kind imposed. This was done intentionally as the Department desired to encourage unrestricted operation.

Gross Weights Are Not in Proportion to Rated Capacities.—Reference has been made to the term "manufacturer's rated capacities." This applies to the single vehicles and the load on the semi-trailer of the semi-trailer combinations.

Fig. 1 indicates the number of motor trucks examined under their different classifications as rated by the manufacturer in various class limits coming within 1000 lb. of weight. It is interesting to note in this chart that the gross loads of the small-capacity vehicles are much greater in proportion to the rated capacity than those of the large-capacity trucks.

Comparing the 1½-ton with the 5-ton class in Table 1, the ratio of rated capacity is 3.33. If the gross rate were proportional to the capacity, 5-ton trucks would have an average gross rate of 3.33 x 10,500 lb., or 35,000 lb.; while, actually, the average was 21,600 lb.

Table 2.—Frequency Distribution of Gross Weights of Loaded Four-Wheel Single-Vehicles, Percentage of Total Observations in Each Capacity Class

Manufacturer's Rated Capacity, Tons	Average Gross Weight, Lb.	Gross Weight, 1000 Lb. 4-6 Per Cent	7-9 Per Cent	10-12 Per Cent	13-15 Per Cent	16-18 Per Cent	19-21 Per Cent	22-24 Per Cent	25-27 Per Cent	28-30 Per Cent	31-33 Per Cent	34-36 Per Cent	37-39 Per Cent	40-42 Per Cent	43-45 Per Cent	Total Observations, Number
1	6,400	70.0	20.0	6.7	.....	3.3	.....	.....	.....	.....	.....	.....	.....	.....	.....	30
1½	10,100	19.8	26.9	23.5	23.5	5.6	0.4	0.1	0.1	0.1	.....	.....	.....	.....	.....	2,695
2	14,300	2.0	14.2	20.1	21.7	22.9	14.8	3.6	0.7	.....	.....	.....	.....	.....	.....	511
2½	16,000	1.4	8.4	15.2	20.5	25.6	14.9	9.8	4.0	0.2	.....	.....	.....	.....	.....	429
3	17,800	0.3	3.3	11.4	15.7	20.4	27.6	15.4	5.3	0.3	0.3	.....	.....	.....	.....	395
3½	18,600	2.8	.....	9.4	17.0	17.9	20.8	18.9	12.3	0.9	.....	.....	.....	.....	.....	106
4	18,800	.....	.....	17.6	5.9	35.2	5.9	11.8	23.6	.....	.....	.....	.....	.....	.....	17
5	20,100	0.5	.....	5.1	7.9	18.1	28.1	28.2	10.7	0.9	0.5	.....	.....	.....	.....	216
Over 5	19,200	4.3	.....	8.7	8.7	17.4	34.9	4.3	17.4	.....	4.3	.....	.....	.....	.....	23

Table 3—Frequency Distribution of Gross Weights of Loaded Six-Wheel Single-Vehicles, Percentage of Total Observations in Each Capacity Class

Manufacturer's Rated Capacity, Tons	Average Gross Weight, Lb.	Gross Weight, 1000 Lb.													Total Observations, Number
		4-6 Per Cent	7-9 Per Cent	10-12 Per Cent	13-15 Per Cent	16-18 Per Cent	19-21 Per Cent	22-24 Per Cent	25-27 Per Cent	28-30 Per Cent	31-33 Per Cent	34-36 Per Cent	37-39 Per Cent	40-42 Per Cent	43-45 Per Cent	
1																0
1½	14,900	2.2	12.9	12.9	25.7	23.1	17.8	4.9	0.5							225
2	17,300		13.7	4.5	9.1	27.3	31.8	9.1		4.5						22
2½	21,200			6.9	13.8	31.0	10.3	6.9	6.9	13.8		3.5	6.9			29
3	22,100		3.1	15.6	6.3	18.8	9.4	6.3	9.4	12.3	6.3	3.1	6.3	3.1		32
3½	29,100				8.0		12.0	12.0	8.0	12.0	8.0	28.0	4.0	4.0	4.0	25
4	24,300		5.6		11.1	11.1	5.6	5.6	22.0	16.7	16.7		5.6			18
5	28,500				2.2	10.9	13.0	13.0	4.3	13.0	6.6	17.4	13.0	6.6		46
Over 5	32,500					5.0	10.0	15.0		5.0	5.0	10.0	30.0	20.0		20

Tables 2 and 3 of these data have segregated the four-wheel and the six-wheel-vehicle classifications, which, as already stated, carried 25,000 lb. and 40,000 lb. respectively.

Table 4 analyzes the heavier gross weights according to rated capacity and shows that, of 4839 vehicles inspected, 427 or 8.9 per cent had a gross weight in excess of 21,000 lb. We ordinarily associate such gross weights with vehicles of 5 ton and over in rated capacity, whereas Table 4 indicates that only 144, or a trifle over one-third of the 427 vehicles with gross weight exceeding 21,000 lb., were in the 5 ton and over class. Over one-fourth of the total number were 2½ tons or smaller, and 16 were of the 1½-ton rated-capacity. It is also interesting to note that only 190 of all loaded vehicles were in the 24,000-lb. gross-weight classification, and that 73 of these were in the 5 ton and over capacity class. It is not until the gross weight rises in excess of 24,000 lb. that the rest of the 1½ and 2-ton trucks disappear. The 61 trucks shown by Table 4 to be in excess of that weight include representatives of all rated-capacity classes from 2½ tons upward; and 0.2 per cent of all the single vehicles weighed above 40,000 lb., and these are rated at 3-ton capacity or more. They are also of the three-axle six-wheel type limited by law to 40,000 lb., which in this case was exceeded; and yet, from a standpoint of road construction, they made less demand on the strength of the road than the heavier four-wheel vehicles in the 21,000-24,000-lb. class. Of the 61 vehicles weighing over 30,000 lb., 58 were six wheelers and were in the 3 ton and over classification.

Fig. 2 shows the distribution for tractor-semi-trailer combinations. The tendency to heavier loading in the smaller units in relation to capacity again prevails. If gross weights were proportional to semi-trailer capacities, the 5-ton class would average 3.33 x 20,800 lb. or 69,200 lb., where the actual average is 27,100 lb. Nearly two-thirds of all the semi-trailer

Table 4—Classification of All Heavy-Loaded Single-Vehicles According to Manufacturer's Rated Capacity and Gross Weight

Manufacturer's Rated Capacity, Tons	Total of all Loaded Vehicles	Loaded Vehicles Having Gross Weights Exceeding:							
		21,000 Lb.		24,000 Lb.		30,000 Lb.		40,000 Lb.	
	No.	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent
1	30								
1½	2,920	16	3.8	3	1.6				
2	533	25	5.9	5	2.6				
2½	458	71	16.6	27	14.2	3	4.9		
3	427	99	23.2	36	19.0	7	11.5	1	10.0
3½	131	54	12.6	31	16.3	12	19.7	2	20.0
4	35	18	4.2	15	7.9	4	6.6		
5	262	121	28.3	54	28.4	21	34.4	3	30.0
Ovre 5	43	23	5.4	19	10.0	14	22.9	4	40.0
Total	4,839	427	100.0	190	100.0	61	100.0	10	100.0
Percentage of Total Loaded Vehicles	100.0	8.9	....	4.0	....	1.2	....	0.2	....

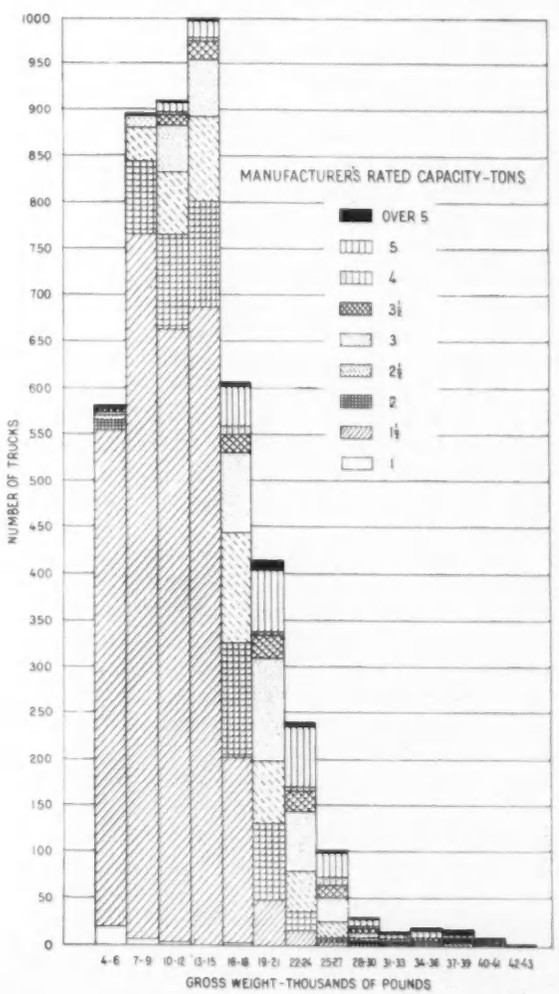


Fig. 1—Frequency Distribution by Gross Weights of Loaded Single Vehicles

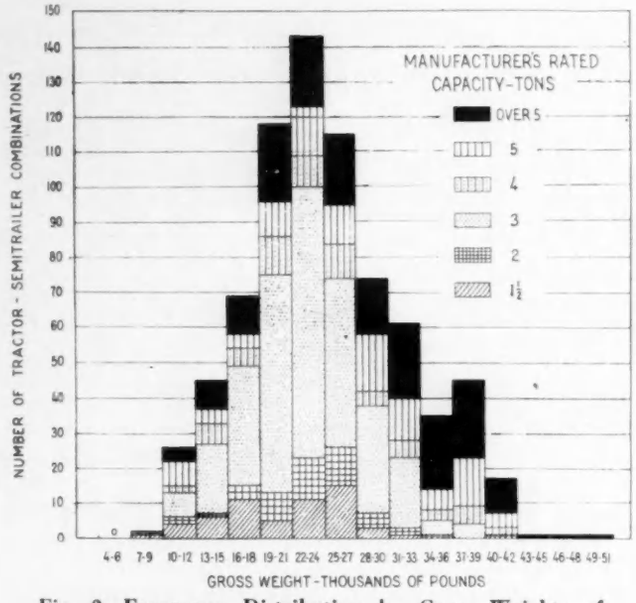


Fig. 2—Frequency Distribution by Gross Weights of Loaded Tractor-Semi-Trailer Combinations

combinations exceed the 21,000-lb. gross-weight; and almost one-half weigh more than 24,000-lb. gross-weight; while one-fifth exceed the 30,000-lb. weight, with 2.7 per cent weighing more than 40,000 lb. This again brings out the fact that the heavier gross loads are not by any means carried exclusively on vehicles of the larger rated-capacities. The smaller-rated vehicles are represented in large number in each group up to the 30,000-lb. limit.

Light Trucks Carry the Greatest Overload.—As previously shown in the analysis of gross weights, it is definitely indicated that loading in excess of rated capacity is greater in the case of small trucks than that of large ones. On the average, the 1½-ton truck will be found to carry from 1 to 1 2/3 times as much; while the 5-ton trucks carry just about their rated capacity. In fact, the smaller trucks are loaded to four, five and even six times their rated capacity; while the larger ones are seldom loaded beyond their rated capacity.

Fig. 3 shows a comparison of the weights carried by the 1½ and 5-ton trucks, respectively, on the basis of the total number of each class in the gross-weight analysis. It is interesting to note that the 1½-ton truck appears on the chart carrying a load between 6 and 10 tons more frequently than does the 5-ton truck.

Rear-Axle Loading.—The data secured from the Survey indicates that, in the case of single vehicles, the weight on the rear axle averages approximately three-fourths of the gross load; that is, above the 10,000-lb. gross-load classification.

In the case of tractor-semi-trailer combinations, 45 per cent of the entire gross weight was carried on the rear ends of both the tractor and the semi-trailer, with about 10 per cent on the front axle of the tractor.

Data on Rear-Wheel Loads Analyzed.—The average rear-wheel load of single vehicles, as shown in Fig. 4, ranged from a minimum of 2220 lb. for trucks of the 1-ton rated-capacity to a maximum of 6910 lb. for 5-ton trucks. Again it is interesting to note that the load on the rear axle of the larger-rated-capacity truck is relatively less than the load on the axle of the smaller-rated-capacity truck. It is also true, especially on the single vehicle, that the greatest wheel loads are not always found on the vehicles of the largest capacity.

In Table 5 (Table 18 of the *Public Roads* Report) the loads exceeding certain limits are classified according to the rated capacity of the vehicle in which they are observed. The limits chosen are 7000, 8000, 9000 and 10,000 lb. It shows that 4705 loaded vehicles were observed and only 465, or less than 10 per cent, had rear-wheel loads in excess of 7000 lb.; that 113, or less than 25 per cent, were of the 5-ton rated-capacity or larger; and that 180, or nearly 39 per cent, were of the 2½-ton capacity or smaller. Vehicles with wheel loads exceeding 8000 lb. number only 212, or 4.5 per cent of the total. It will again be noted that the number of vehicles in the 5 ton and over classification are less in number than the 2½ ton and under capacity; that 60, or 1.3 per cent, of the total number of vehicles had wheel loads exceeding 9000 lb.; and that only 15, or about 0.3 per cent, had wheel loads greater than 10,000 lb. In these two groups, vehicles of larger capacity were exceeded by the number of vehicles in the 2½-ton-capacity class and smaller.

Fig. 5 indicates the wheel load of the loaded tractor, showing the manufacturer's rated capacity of trailer tons, the number of tractors in the Survey and the load on the rear wheels in thousands of pounds.

Table 5 (Table 18 of the *Public Roads* Report)—Classification of All Loaded Single Vehicles with Heavy Wheel-Loads According to Manufacturer's Rated Capacity and Rear-Wheel Loads

Manufacturer's Rated Capacity Tons	Total of all Loaded Vehicles	Rear-Wheel Loads Exceeding:							
		7,000 Lb.		8,000 Lb.		9,000 Lb.		10,000 Lb.	
	No.	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent
1	41	.....	.....	.....	.....	.....	.....	.....	.....
1½	2,841	13	2.8	4	1.9	2	3.3	.....	.....
2	515	72	15.5	17	8.0	3	5.0	.....	.....
2½	460	95	20.4	53	25.0	14	23.3	5	33.3
3	403	119	25.6	53	25.0	18	30.0	3	20.0
3½	129	45	9.7	24	11.3	9	15.0	3	20.0
4	37	8	1.7	5	2.4	1	1.7	1	6.7
5	243	101	21.7	54	25.5	12	20.0	3	20.0
Over 5	36	12	2.6	2	0.9	1	1.7	.....	.....
Total	4,705	465	100.0	212	100.0	60	100.0	15	100.0
Percentage of Total Loaded Vehicles	100.0	9.9	....	4.5	....	1.3	....	0.3	....

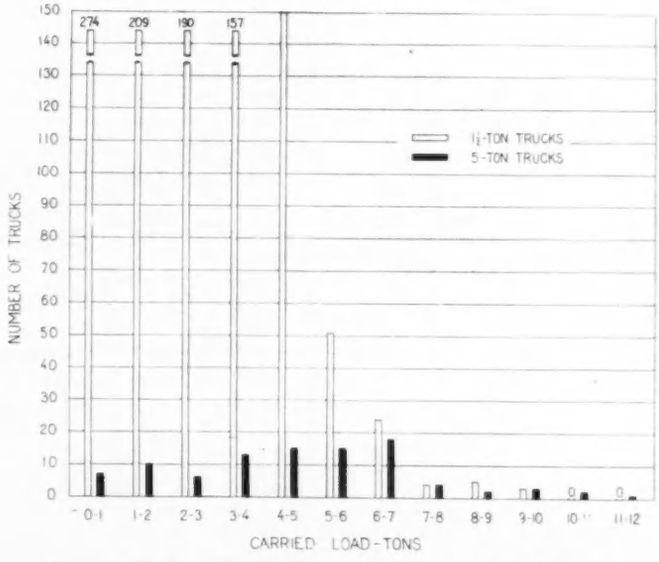


Fig. 3—Trucks of 1½ Ton and 5-Ton Rated Capacity Classed According to Loads Carried

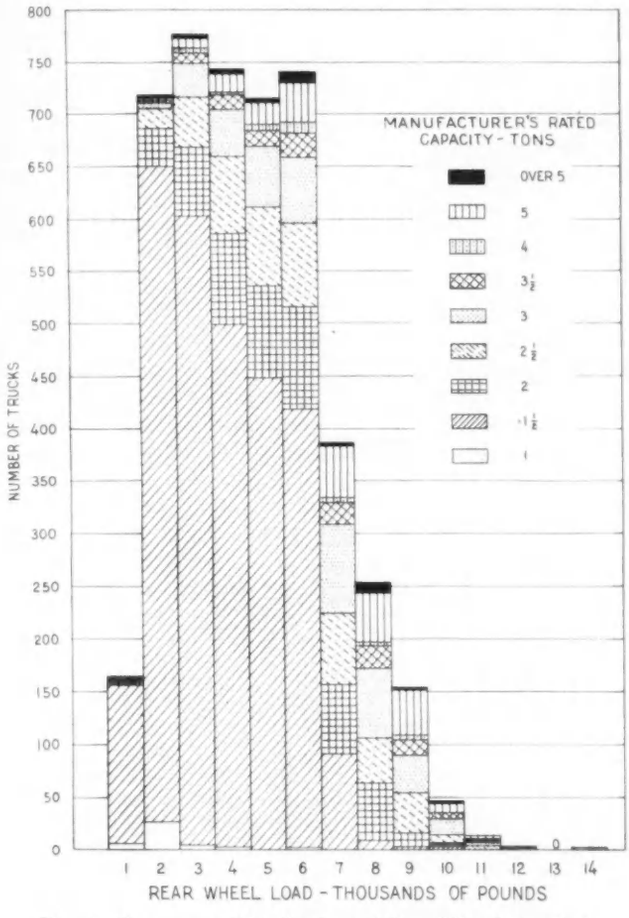


Fig. 4—Frequency Distribution of Rear-Wheel Loads for Loaded Single Vehicles

Fig. 6 indicates the number of semi-trailers in this Survey, the rated capacity in tons and the load on the rear axle in thousands of pounds.

Table 6 (Table 19 of the *Public Roads* Report) indicates the rear-wheel loading exceeding the 7000, 8000, 9000 and 10,000 lb. of the loaded tractors, showing that the rear-wheel loads on the tractor and semi-trailer are greater on the larger-capacity vehicles than on those of the smaller.

This is also true of the Survey made of rear wheels of loaded semi-trailers, as shown in Table 7 (Table 20 of the *Public Roads* Report).

It is evident that excessive overloading is confined primarily to trucks in the lower-rated-capacity class.

The industry uses a formula for rating the ability of a motor truck as an indication of its capacity; that is, 0.045 ability factor in high gear, using the Myers formula:

$$\text{Ability Factor} = \frac{T \times G.R. \times Eff.}{R.R. \times G.V.W.}$$

where

T = Torque
G.R. = Gear Ratio
Eff. = Efficiency; 0.90 in direct; 0.85 in gear
R.R. = Rolling Radius of Tire
G.V.W. = Gross Vehicle Weight

It will be noted that this does not take into consideration acceleration or speed and is not satisfactory from that standpoint.

The National Conference on Street and Highway Safety, at its meeting in Washington in May, 1934, had this subject under discussion. A proposal was adopted in the form of a suggestion to State legislatures to require that all motor trucks be loaded only to the extent that the truck be able to ascend a 3-per cent grade in the level country and a 6-per cent grade in the hilly districts at 20 m.p.h.

The State of Rhode Island highway officials have tried to enforce a regulation requiring trucks to have the ability to ascend a 4-per cent grade at 20 m.p.h. This is considered by many operators and manufacturers to be a severe requirement, and yet it does not cure the evil of *overloading*.

Inasmuch as the subject of rating, loading, ability, and the like, are to be given a great deal of attention by state and highway officials in the near future, the Society, through the Transportation and Maintenance Committee, has been studying the seriousness of the situation together with the Rating Committee "for the purpose of setting a performance rating acceptable to the industry, sufficiently accurate to be of best service to users and fair and simple enough to form a safe basis for legislation or regulation, if the latter must come."

The following formula was set up for consideration:

$$\text{Performance Factor} = \frac{3.34 \times \text{Piston Displacement} \times \text{R.P.M.}}{\text{Gross Vehicle Weight}}$$

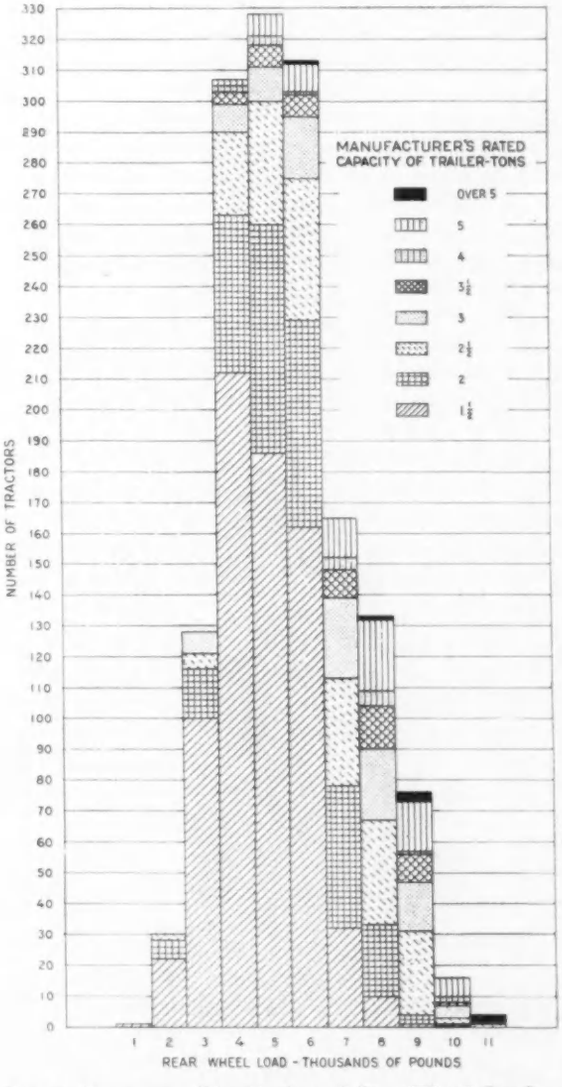


Fig. 5—Frequency Distribution of Rear-Wheel Loads of Loaded Tractors

Table 6 (Table 19 of the *Public Roads* Report)—Classification of All Loaded Tractors with Heavy Wheel-Loads According to Manufacturer's Rated Capacity and Rear-Wheel Loads

Manufacturer's Rated Capacity Tons	Total of all Loaded Vehicles	Rear-Wheel Loads Exceeding:							
		7,000 Lb.		8,000 Lb.		9,000 Lb.		10,000 Lb.	
	No.	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent
1½	726	11	4.8	1	1.0	.....	.....	.....	.....
2	287	27	11.8	4	4.2	1	5.0	.....	.....
2½	209	64	27.9	30	31.3	3	15.0	1	25.0
3	116	43	18.8	20	20.8	4	20.0	.....	.....
3½	51	24	10.5	10	10.4	1	5.0	.....	.....
4	18	8	3.5	3	3.1	2	10.0	.....	.....
5	76	45	19.6	22	22.9	6	30.0	.....	.....
Over 5	8	7	3.1	6	6.3	3	15.0	3	75.0
Total	1,491	229	100.0	96	100.0	20	100.0	4	100.0
Percentage of Total Loaded Vehicles	100.0	15.4	....	6.4	....	1.3	....	0.3	....

The complete report is published in the September, 1935, issue of the *Commercial Car Journal*, page 40.

Inasmuch as the engine r.p.m. is a vital factor in this formula, the shift is from torque to horsepower, and therefore does not accomplish what is desired in the overloading of a light-weight truck or tractor-semi-trailer; for example:

A truck or tractor-semi-trailer of 24,000-lb. capacity equipped with a 215-cu. in. displacement engine having a maximum of 76 hp. at 3800 r.p.m. will have a performance factor of 113.7. A similar-capacity truck or tractor-semi-trailer with a 341-cu. in. displacement engine with a maximum of 76 hp. at 2400 r.p.m. will have the same performance factor of 113.7, due to the difference in speed only. The wear factor will be about 58 per cent greater in the small engine. It will ascend a 4-per cent grade at 20 m.p.h., using a 9.75—22 tire with a 22.5:1 total ratio in second speed in the transmission in the small truck, and 14.2:1 total ratio in third gear in the larger truck, on a 25-lb. per ton road.

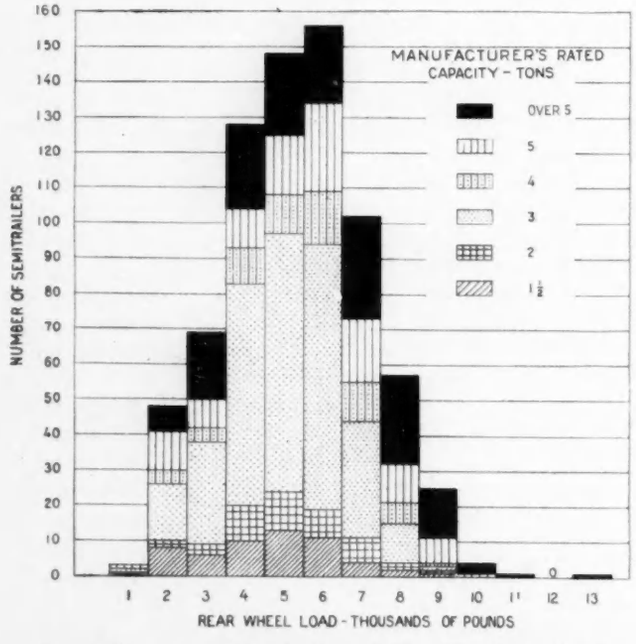


Fig. 6—Frequency Distribution of Rear-Wheel Loads of Loaded Semi-Trailers

To state the problem another way, it will require 76 hp. to ascend a 4-per cent grade at 20 m.p.h. regardless of gear ratio or tire size used.

Acceleration, Deceleration, and Safety.—A light-weight motor-truck equipped with a small high-speed engine running about 3600 to 3800 r.p.m. maximum speed will have the same acceleration that a larger truck equipped with a larger engine having a maximum horsepower at the lower speeds, provided that both are equal from a load and a mechanical standpoint.

The lighter lower-priced truck does not have the brake equipment to get satisfactory deceleration; that is, the diameters of the brakes are usually smaller, requiring greater pressure per square inch and less width, having a reduced wear factor, requiring more frequent adjustment and servicing due to having about 40 per cent less total square inches of braking area than is used on a truck in the higher-priced greater-capacity class. This is a safety hazard and is something that the operator cannot afford to overlook.

Table 7 (Table 20 of the *Public Roads* Report)—Classification of All Loaded Semi-Trailers with Heavy Wheel-Loads According to Manufacturer's Rated Capacity and Rear-Wheel Load

Manufacturer's Rated Capacity Tons	Total of all Loaded Vehicles	Rear-Wheel Loads Exceeding:							
		7,000 Lb.		8,000 Lb.		9,000 Lb.		10,000 Lb.	
	No.	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent
1½	55	3	3.4	1	3.2	.....	.....	.....	.....
2	45	3	3.4	1	3.2	.....	.....	.....	.....
3	302	12	13.6	1	3.2	.....	.....	.....	.....
4	62	7	8.0	1	3.2	.....	.....	.....	.....
5	110	19	21.6	8	25.8	1	16.7	.....	.....
Over 5	168	44	50.0	19	61.4	5	83.3	2	100.0
Total	742	88	100.0	31	100.0	6	100.0	2	100.0
Percentage of Total Loaded Vehicles	100.0	11.9	....	4.2	....	0.8	....	0.3	....

Economical Loading.—It is evidently difficult to evaluate a motor truck by the Gross-Rating method only, as the rating varies with every manufacturer's system of rating. The tonnage rating is primarily for tax purposes. A true measure of the loading value of a motor truck is made by analyzing the units used in the manufacture of the truck; that is, (*a*) chassis weight, (*b*) axles, (*c*) engine, (*d*) transmission, (*e*) clutches, (*f*) frame, and (*g*) tires.

The chassis weight will, as a measure of value, be attacked by some as being of no value due to the fact that it does not require engineering skill to make a truck heavy; but it does require a genius to design a machine that will do the work required with the least possible weight. This is granted. When we refer to chassis weight, we do not mean putting lead in the wheels or pig iron under the frame structure. When the different units are made with the desirable factor of safety that automatically fixes the weight when using the finest materials. A comparison made of the different chassis in the field, and a comparison of the different units in the chassis that are comparable, will show a very little difference in chassis weight.

Compare the rated capacity of the *bearings* in the axles. The higher the rating, the heavier the axle. Axles having the same rated capacity will weigh about the same. *Engines* of the same cubic-inch displacement weigh about the same.

Transmission and *clutches* having a 10 to 15 per cent input-torque capacity in excess of the engine will weigh about the same. *Frames* that will be suitable for the foregoing axle, engine, and so on, will weigh about the same. *Tires, wheels* and *springs* large enough to carry the load will weigh about the same. Summing it all together will provide a very good rule to use as a measure of carrying ability; that is, *chassis weight*.

When the foregoing is taken into consideration in the purchase of a motor truck, the price will vary only a few dollars. Ultimate cost is then more important.

First Cost.—This, as shown in Fig. 7, is only about 20 per cent of the total cost of owning a motor truck. Frequently, it receives the major consideration. The 80 per cent of the truck expense is about the same for trucks of the same size.

Fixed Cost.—Including license, insurance, rent, and the like; about the same for all trucks.

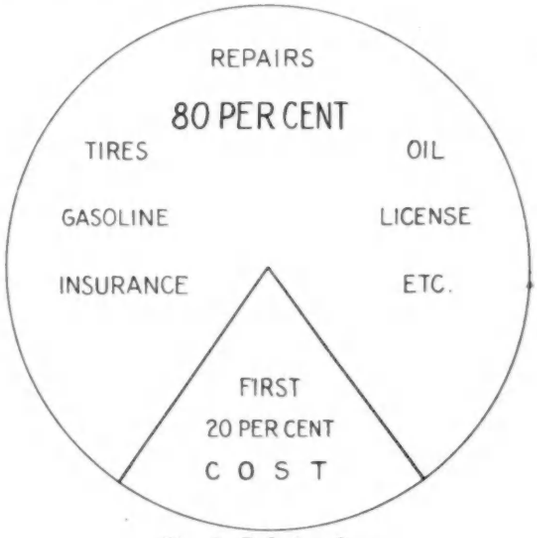


Fig. 7—Relative Costs

Gasoline, Oil, and Tires.—Costs vary with load and road conditions with trucks of the same specifications.

Depreciation.—This will be less on a higher-priced quality-truck than on a cheaper overloaded truck.

A Difference in First Cost of	With a Life of	For the Same Depreciation, a Higher-Priced Truck Will Be Off the Books
$100	5 years, or 100,000 miles	5 months later, or 8,000 miles more
$200	4½ years, or 90,000 miles	10 months later, or 13,000 miles more
$300	3½ years, or 70,000 miles	12 months later, or 20,000 miles more

Many records prove that the economical life is greater than that shown by the chart before the obsolescence period has been reached.

An analysis of hundreds of cost records has shown that the higher-priced trucks doing the same or more work than a cheaper-priced truck will operate for ½ to 1½ cents less per mile of operation.

To illustrate, one department store using the high and the low-priced trucks in the same kind of work in an Eastern city analyzed its cost records covering a period of 60,000 miles for the low-priced trucks compared with the 100,000-mile records of the higher-priced trucks. The complete analyses of the repair costs of the groups of trucks are shown in the curves plotted in Fig. 8.

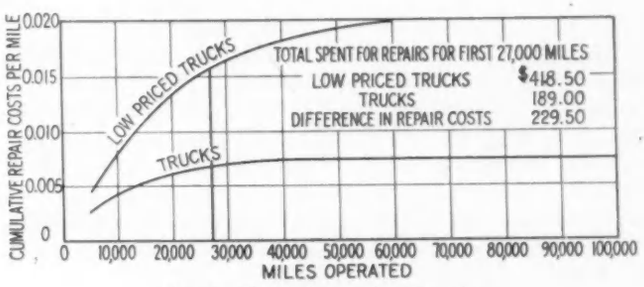


Fig. 8—Total Spent for Repairs

A Difference in First Cost of	Will Be Offset, Miles	Saving in Repair Costs, Cents Per Mile
$200	20,000	at 1
	36,700	at ¾
	40,000	at ½
$300	20,000	at 1½
	30,000	at 1
	40,000	at ¾
	60,000	at ½
$400	26,700	at 1½
	40,000	at 1
	53,300	at ¾
$500	33,300	at 1½
	50,000	at 1

This analysis shows that, in 27,000 miles of operation, the low-priced trucks cost $229.50 more per truck for repairs than did the higher-priced trucks at the same mileage. This amount practically equalled the difference in first cost of the two groups of trucks.

At 27,000 miles, the total repair costs per mile for the low-priced trucks were over twice the costs for higher-priced trucks and continued to increase until the 60,000-mile period; while the repair cost per mile for the higher-priced trucks remained about the same.

At the end of 60,000 miles, the total repair cost per truck, for the low-priced trucks, was $1,170. The total repair cost, per unit, for the higher-priced trucks was $450, or a total saving in repair cost alone at the end of the 60,000 miles of $720. This is equal to *three times the difference in first cost.*

This low-priced truck probably will have performed satisfactorily in its own class. The rapid increase in repair costs for the lower-priced trucks is the reason that it is uneconomi-

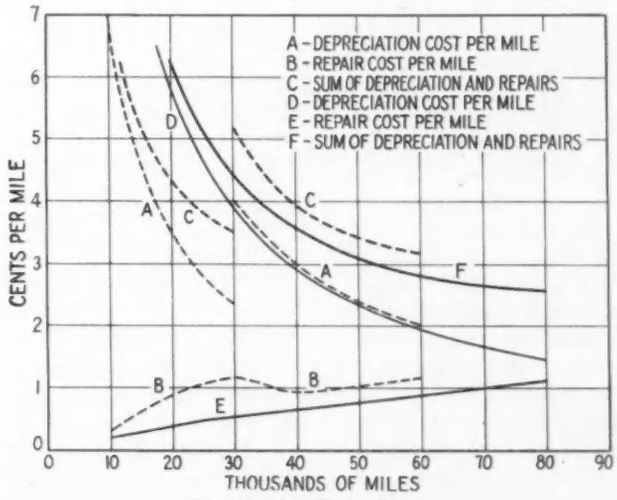


Fig. 9—Cost-Data Curves

cal to operate in a field to which these trucks were not suited, or an overloaded condition.

The dotted curves in Fig. 9, are: (*A*) Depreciation, (*B*) repair cost, and (*C*) total depreciation and repair costs per mile on a low-priced truck; (*D*) is depreciation, (*E*) repair cost, and (*F*) total depreciation and repair costs per mile on a higher-priced truck.

Curve (*A*) is the depreciation cost, showing 0.07 cents per mile for the first 10,000 miles and 0.0235 cents per mile at 30,000 miles.

Curve (*B*) is the repair cost, showing 0.003 cents for the 10,000 miles and 0.0115 cents per mile at 30,000 miles. The truck was traded-in at this period and a new truck purchased.

The new truck was put on the books at 30,000 miles with a depreciation cost of 0.04 cents per mile, which represents the difference between the purchase price and the trade-in value. The repair costs for the next 10,000 miles were lower, due to low repair cost per mile for the new truck in this period. They changed from 0.0115 cents per mile at 30,000 miles to 0.009 cents per mile at 40,000 miles. At 40,000 miles the repair cost per mile increased from 0.0093 cents per mile up to 0.0117 cents per mile at 60,000 miles.

Curve (*C*) is the combined depreciation and repair cost for each mile, or 0.0517 cents per mile at 30,000 miles to 0.0315 cents per mile at 60,000 miles.

Curve (*D*) is the depreciation cost per mile, showing 0.065 cents per mile at 18,000 miles and 0.0145 cents per mile at 80,000 miles.

Curve (*E*) is the repair cost, showing 0.00525 cents per mile at 30,000 miles and 0.011 cents per mile at 80,000 miles.

Curve (*F*) is the combined depreciation and repair cost, being 0.0625 cents per mile at 20,000 miles and 0.0255 cents per mile at 80,000 miles.

Thus showing a saving of 0.006 cents per mile with 20,000 miles additional operation and still going down in operating costs without approaching the period of obsolescence.

In making a comparison of the operation of two trucks, the question will be raised as to why the cost of operating the first truck was considered in the operation of the second truck. It is necessary to do this because a comparison is being made between two types of vehicles, and, if at the end of 60,000 miles a third vehicle should be purchased, the cost of the previous two trucks should be included in order to give a true comparison. If a comparison is made of the operating costs of two trucks of the same carrying capacity and same price, then there would be no point in accumulating the cost because both of the vehicles would be in the same class.

We are firmly of the opinion that there is no economy in the purchase or use of a low-capacity low-priced truck, heavily overloaded, in comparison with a truck of higher price and carrying capacity, when all of the factors are taken into consideration.

Discussion of Scaife Paper

Hope Expressed for a Standard Rating Chart

—L. V. Newton

Byllesby Engineering and Management Corp.

THE industry is without a yardstick with which to rate trucks as to carrying capacity and the operator must rely solely upon the manufacturer for his recommendations as to the allowable gross-vehicle weight for a given vehicle. Of course, we rate a truck's gross allowable load by its tire size; but, again the maximum tire size is dictated by the truck manufacturer.

I hope that a standard rating chart may be evolved by the Society and that it will consider the capacities under normal and shock load of axles, springs, spindles, axle shaft and frames; also, horsepower requirements should be set forth to meet reasonable ability requirements.

Constructive Criticism on Important Subjects

—Fred B. Lautzenhiser

International Harvester Co.

IT is a deplorable fact, as pointed out by Mr. Scaife, that tonnage ratings as used today are in no sense indexes to the loads that trucks could or should or do carry. Nor do tonnage ratings bear any relation to the so-termed gross-ratings as they are used. In my opinion Mr. Scaife has told practically the whole story of the Maryland Survey in his second paragraph, and has emphasized it with his *Commercial Car Journal* tabulation in proof of his statement. The Maryland Survey overlooked the fact that the tonnage rating is not—to quote from the original article in the *Public Roads* Report—"the manufacturers' recommended loading", etc.

The gross-rating method, although possibly being a step in the right direction, stops short of any real value other than that of an index to the size of tires required, assuming that the distribution of gross between front and rear is known. Some very interesting facts can be found by a study of Mr. Scaife's Maryland Survey charts and tables.

First, the average gross weights checked in the Maryland Survey are invariably less than the averages of the grosses for the various classifications tabulated from the *Commercial Car Journal*, as shown in Table *A*.

The tabulation of gross-vehicle weights taken from the *Commercial Car Journal* for July, and which Mr. Scaife arbitrarily calls "gross ratings", are assumed to be correct although, even with a rather diligent search, I could locate but a few. Be that as it may, it is a fact nevertheless that these average loads in every classification are generally well below the gross-weight allowances for International models in the same tonnage classifications, these being the trucks with which I am most familiar.

This lends the belief that, generally speaking, we all may be needlessly alarmed at the much-discussed tendency toward overloading. There are many specific cases of overloading that are almost unbelievable; however, the Survey shows these cases to be negligible and an extremely small percentage.

Early in Mr. Scaife's paper he states: "In order to get a true picture of operating conditions, no attention was paid to violation of these laws (Maryland), nor were any penalties of any kind imposed." This statement is slightly in error as page 38 of the May, 1935, issue of *Public Roads* states: "After the middle of October, at the insistence of State authorities, the laws were enforced and nine arrests were made for overweight, 1500 weighings being recorded during the corresponding period." This represents but 0.6 per cent

of all trucks weighed during that period, which certainly speaks well for law observance among truckers in Maryland.

Looking over Table 3, covering the six-wheel units, it is noted that 54 per cent of all these observed are shown as being rated by the manufacturers at 1½ tons and as having an average gross weight of 14,900 lb. each. It is significant to note that the Motor Truck Bible, the *Commercial Car Journal*, does not list a single 1½-ton six-wheel truck. No manufacturer makes such an animal, so we must assume these to be four-wheel 1½-ton units with some sort of a makeshift third axle attached, and for which finally complete unit no manufacturer can or will be responsible for or stand behind. Here, in my opinion, is a hazard to everyone on the highway.

Mr. Scaife gave a mathematical formula, the calculation of which results in a so-called "Ability Factor." This ability factor is nothing more nor less than the tractive effort in pounds that is available for each pound of gross-vehicle weight. He further stated that "this does not take into consideration acceleration or speed and is not satisfactory from that standpoint." For calculation of performance ability, we find a modification of the formula very satisfactory and have used it for many years.

The road speed involved in miles per hour is also readily calculated, using the same factors of gear ratio and effective tire-diameter simply by using the speed of the engine at which the torque value used was developed.

I have no great admiration for the performance-factor formula given, in which the factor equals the constant value 3.34 multiplied by the piston displacement, multiplied by the engine speed, and divided by the gross-vehicle weight. Due to the facts that this formula injects into the picture a constant value the origin of which is unknown to the general public, that the speed of the engine is also injected into the formula, and that the formula itself departs from the use of the engine torque-output and injects the horsepower factor into the picture, all possible interest of mine is thus eliminated.

I am inclined to ride along with the State of Rhode Island in its requirement for all vehicles or combinations to be able to climb a 4-per cent grade at a speed of 20 m.p.h. That would unquestionably cure the evil of overloading if it were enforced. Mr. Scaife stated that it did not cure the overloading evil. True, it did not, for the simple reason that no effort was made to enforce it because of the wails and howls of anguish from manufacturer and user of trucks alike.

The injection of involved mathematical formulas or other more or less technical specifications will never serve any good purpose when it is expected that the general public must be more or less familiar with them. The general public does not care by what means a result is arrived at, but is directly interested in the result itself.

I cannot in any way subscribe to Mr. Scaife's remarks on "Acceleration, Deceleration and Safety", because the performance ability of a light-weight motor-truck with a small high-speed engine is in no way comparable with that of a larger truck equipped with a larger engine. In the first place, they would not intelligently be loaded the same and they would not have the same tire size nor axle ratios; but, assume them each to be properly loaded, and they should have approximately the same deceleration rate, brake life and safety factors.

At the point where Mr. Scaife starts to discuss operation costs, he becomes so bewildering that I hardly know what comments to offer. We do agree thoroughly in that a light-duty truck should not be used in heavy-duty work, nor a heavy-duty truck in light-duty work. The nature of the operation itself should be studied carefully and the proper unit put in that will satisfy the requirements.

There are only five major factors to satisfy in any transport operation in order to result in an efficient operation; they are: (*a*) maximum load permitted, (*b*) maximum speed permitted, (*c*) continuous operation, (*d*) lowest operating cost, and (*e*) safety.

More or less summarizing a study of the Maryland Survey and Mr. Scaife's paper in general, and injecting the few remarks which are believed to be pertinent to the subject, in our opinion tonnage ratings of motor trucks as applied today neither serve nor purport to serve any useful purpose other than that of a general classification for establishing license fees in certain States. It appears that the Maryland Survey completely muffed this fact at the start, as they repeatedly and erroneously refer to the tonnage rating as the capacity of the unit.

Gross ratings, or straight ratings, as they are sometimes termed, are unsatisfactory yardsticks to follow in the sale or purchase of motor trucks; because, being essentially based on the tire equipment of a given unit, either one of two conditions apparently must exist, that is, if equipped with the smallest approved tires a low gross-rating obtains, and the purchaser is being forced to buy too much truck, or, if equipped with the largest approved tires a high gross-rating obtains and he is not buying enough truck.

It appears that the most sensible manner in which to rate a truck for capacity alone is either in pounds or, preferably, tons of load the standard chassis is capable of carrying in the combined weights of cab, body, payload and equipment. This is definitely pre-determined by the manufacturer when designing the various load-carrying members. If the body and equipment is unduly heavy, naturally the payload must suffer; while the opposite holds true, with a light body and equipment the payload automatically increases.

For the purpose of establishing license fees on a basis equitable to both State and operator, a different picture is

Table *A* (Lautzenhiser Discussion)

Commercial Car Journal Tonnage and Gross Ratings on Four-Wheel Trucks			Maryland Survey Four-Wheel Units; Table 2, Scaife Paper
Tonnage	From: Gross Weight, Lb.	Average Gross Weight, Lb.	Average Gross Weight, Lb.
1	7,100 to 11,000	9,050	6,400
1½[a]	6,000 to 12,500	9,250 11,500 Ford; Chevrolet; Dodge	10,100
2	11,400 to 19,000	15,200	14,300
2½	13,000 to 26,000	19,500	16,000
3	16,300 to 32,000	24,150	17,800
3½	19,000 to 25,000	22,000	18,600
4	19,000 to 38,000	28,500	18,800
5	23,000 to 37,000	30,000	20,100

[a]This means that the averages of all four-wheel-unit loads observed and shown in Table 2 of Mr. Scaife's paper were less than the average allowances of the various manufacturers of current trucks in those particular classifications.

presented. The actual wheel weights and/or gross weight involved in operation on and in damage to the highway should indicate the amount, increasing by 500 or 1000-lb. increments.

We agree with Mr. Scaife to a certain extent in that a chassis of a given classification having appreciably heavier load-carrying members—frame, springs, axles and wheels—unquestionably has the greater carrying capacity regardless of the materials used, because the possibilities for weight saving are so limited in those few units as to be negligible.

Overloading of four-wheel units beyond manufacturers allowances as shown by the Maryland Survey is certainly not as prevalent as generally believed. There are practically no law violations in Maryland from overloading.

The equal or possibly greater importance of individual axle or of wheel loads to or over that of gross weight has been illustrated many times, originally in the same publication, *Public Roads*. Tests show that the stresses in rigid pavements caused by vehicular impact loads are a function of wheel load rather than of gross load. Quoting from the publication, "If axles are spaced no closer than 3 ft. between centers, it has been determined that the maximum stress caused by the combined wheel loads does not exceed that caused by each wheel load separately. As wheel spacing closer than 3 ft. is impracticable, the maximum wheel load of vehicles is the critical factor in design of pavement and it is therefore the wheel load rather than the gross load that must be limited for pavement protection."

Continuing the quotation with some omissions: "For bridge protection, limitation of the wheel load is not sufficient, limitation of gross load is also required, and the form of limitation must give proper recognition to the length over which the load is applied."

"For the purpose of limiting the gross weight, and the like, the American Association of Highway Officials has recommended the use of the formula, $W = C\ (L + 40)$ in which W is the gross weight, C is a coefficient and L is the length in feet between the centers of the extreme forward and rear axles of the vehicle or combination. The Association recommended a value of 700 as the lowest value of C to be adopted as a limit in any State. Also it is intended that the gross-load limit given by the formula be used in conjunction with an axle-load or wheel-load limit." These wheel-load limits are 8000 lb. for high-pressure pneumatic and 9000 lb. for low-pressure pneumatic-tires. Naturally, doubling these maximum wheel loads gives us the maximum axle loads of 16,000 lb. and 18,000 lb. for high-pressure and low-pressure tires respectively.

In closing, I might point out four things that the industry, both manufacturer and operator, and the State as well, need:

(1) A uniform and simple method of rating trucks for capacity, free from involved formulas, and the like.

(2) A uniform method of licensing trucks in all the States.

(3) Uniform performance requirements, hill-climing speeds and rates of deceleration in all the States.

(4) Uniform restrictions on weights and dimensions.

Mr. Lautzenhiser's Criticisms Commented Upon by Mr. Scaife

WHILE it is true that the Maryland Survey used average gross weight, the average was not taken by using the lowest and highest and dividing by two as did Mr. Lautzenhiser; but the total number of trucks in the Survey was taken and the average of the total was found, which accounts for the low average in the Survey inasmuch as there were more trucks with an average or light load than trucks overloaded.

However, I was not considering the average load of trucks in each division, but the actual load carried by the trucks. One cannot use averages for overloading any more than one can use averages for speeding. It would be very nice if we could get our Enforcing Officers to consider the average speed of the cars that pass over a given route during the day instead of picking on a few of us that happen to be going rather fast.

While the percentage of overloaded trucks is small in each group, the light or 1½-ton chassis was the greatest offender in every group, wherein it should have been the least. It is also the few trucks that cause trouble for the many operating correctly.

It is true that the violations were enforced from the middle of October to the middle of November, during which period about 1500 trucks were checked. I was not interested in this period. I was interested in the Survey from June to the middle of October before the penalties were put into force, as the Survey became of no value when these penalties were enforced due to the fact that the operators would either load correctly or stay away from this vicinity, for it became a trap. This is the reason why there were very few violations reported during that one-month period.

With reference to the Rhode Island regulation, it failed not only from the "wails of anguish" but it was of no value because it was possible for some of the lighter trucks to comply with the regulations by increasing the speed of the small engine up to 3600 to 3800 r.p.m., thereby carrying the load up a 4-per cent grade at 20 m.p.h., as indicated by the example given in my paper in the paragraph beginning: "A truck or tractor-semi-trailer of 24,000-lb. capacity . . .", and so on.

With reference to the acceleration, deceleration, and safety, I did not compare a light high-speed engine with a heavy low-speed engine. I compared horsepowers. A light high-speed engine developing a maximum of 76 hp. will have exactly the same acceleration as a larger slow-speed engine having the same maximum horsepower, provided both vehicles are loaded the same and have the same efficiency. This is an actual scientific fact, and arguments to the contrary are of no avail. However, the deceleration and safety of these light overloaded vehicles cannot be compared with a heavier vehicle carrying the same load and having greater braking area, which tends to greater safety.

It is unfortunate that Mr. Lautzenhiser becomes bewildered on the discussion of operating costs, as we are simply comparing the operating costs of two vehicles in the same service, one of which was a low-capacity vehicle compared with a high-capacity vehicle carrying the same load, and the records are those of the operators and are actual illustrations.

With reference to the different formulas referred to throughout the paper, it is true that they were not intended to indicate when a truck was overloaded and that they undoubtedly served the purpose for which they were intended. I tried to make it clear that, to date, a yardstick that can be used for indicating when a truck is overloaded has not been discovered in the nature of a formula. We were suggesting a simple method, and that was to use a chassis weight as a base, comparing the size of the units in that particular chassis together with the carrying capacity of the bearings in the different wheels, for checking an overloaded vehicle, as I am fully aware of the fact that complicated and technical formulas are of no value as a regulating guide.

Bringing Italian Skies Down to the Earth

By M. E. Merriman

Technical and Commercial Representative in America for the Fiat Co.

THE Fiat high-altitude aircraft-engine testing-plant can duplicate quickly, conveniently and accurately any combination of atmospheric conditions that may be found in an imaginary vertical column of air rising from sea-level to more than 6 miles. To compare results, the output of an engine is measured in "standard air", which is characterized by a definite pressure, temperature and humidity.

The high-altitude testing-plant comprises the following:

(1) An engine test-cell, with a volume of approximately 6000 cu. ft., which is thermally insulated and airtight.

(2) The refrigerating plant, which comprises a refrigerating machine, the cooling surfaces which absorb heat from the air enclosed in the test-chamber, the air fed to the carburetor of the engine, and the exhaust gases flowing from the engine.

(3) An air-pressure-reduction plant which includes a turbo-blower, an automatic altitude-regulator and an exhaust-gas cooler.

(4) The artificial-wind plant which can supply up to a wind speed of 100 m.p.h. at a pressure corresponding to an altitude of 16,300 ft.

(5) The apparatus for humidifying the engine intake-air.

(6) The engine-test apparatus; hydraulic and electric power-brakes are used. The apparatus makes it possible to check the oil and the fuel consumption, and the water temperature at the inlet and outlet of the engine.

(7) The central control-board was laid out so that the operator always has the entire test-plant under full observation and control.

The Fiat high-altitude testing-plant is the result of a long series of experiments and research work.

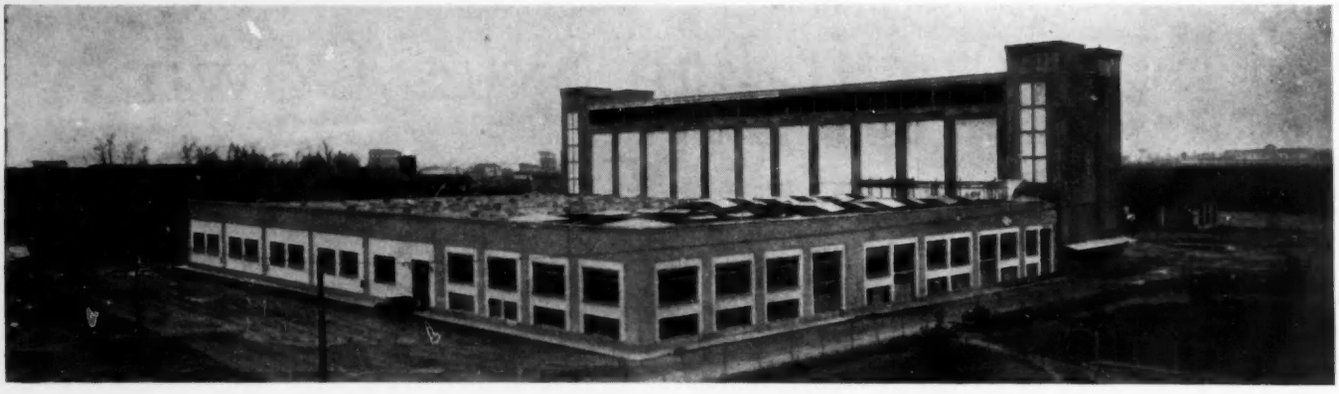
IN every age, for centuries past, there have been those who have dreamed of soaring birdlike over neighboring hills and valleys, high above the heads of their fellow men. And these dreams have led to more or less serious though futile attempts to realize this, until of late years, most fantastic of notions. Several of the more heroic of these attempts have been recorded from time to time in the pages of history, and have thus become new points of departure of the dreamers of subsequent generations, each adding its mite to the slowly accumulating fund of pre-elementary flying-experience. The immediate object was always the same, simply to get up into the air to become a human bird, the ultimate of human conception beyond which the imagination had not traveled. What the nature of their surroundings would be, once they were in the air, did not interest them.

For example, the effect of altitude on the characteristics of the suspending fluid, air, was not even recognized, much less was it a thing to be at all concerned about. Today we know that there is a lot more to the problem than simply soaring up into space. We now recognize many problems not even dreamed about a few years ago, the whole field is more inclusive, and the subject is more complex, and more delicate.

We now know that the design of airplane engines must contemplate the effects of altitude and that general layout, dimensions, materials of construction, the cooling, fuel, and lubrication systems, et cetera, must all be designed so as to function perfectly at *all* altitudes, in all seasons, in all kinds of weather, under all conditions of use, and with varying degrees of skill in operation. Appropriate regard for human life has made it imperative that we know beforehand how the finished machine will behave under these countless possible combinations of conditions. Therefore, adequate means for carrying on experiments and tests under conditions identical with those in which the completed engine will be called upon to operate, are indispensable to the designer.

This is particularly necessary in the case of engines in-

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 17, 1935.]



The Fiat Aircraft-Engine Test-Plant

The High-Altitude Test-Plant Is in the Low Building in the Foreground, the Silent Test-Plant Being in the Tall Building at the Rear

tended for high-altitude work, and while there is no particular difficulty is designing a test-room that will simulate flying conditions at high altitudes, the cost of construction of such a laboratory has retarded the availability to engineers of experimental equipment of this kind. The problem is not to build a laboratory for the testing of engines on some high mountain; this would serve no useful purpose, even though a conveniently located mountain were available. Such a laboratory would serve only insofar as the behavior of the engine at that particular altitude and under the then existing atmospheric conditions might be concerned, and would of course have no value whatever as a means for determining engine characteristics or for studying requirements of design under conditions of higher or lower altitudes.

The true conception of the atmospheric conditions which would have to be duplicated in an adequate laboratory would be to consider a vertical column of air, rising from sea level to a height of something over 6 miles, under all possible

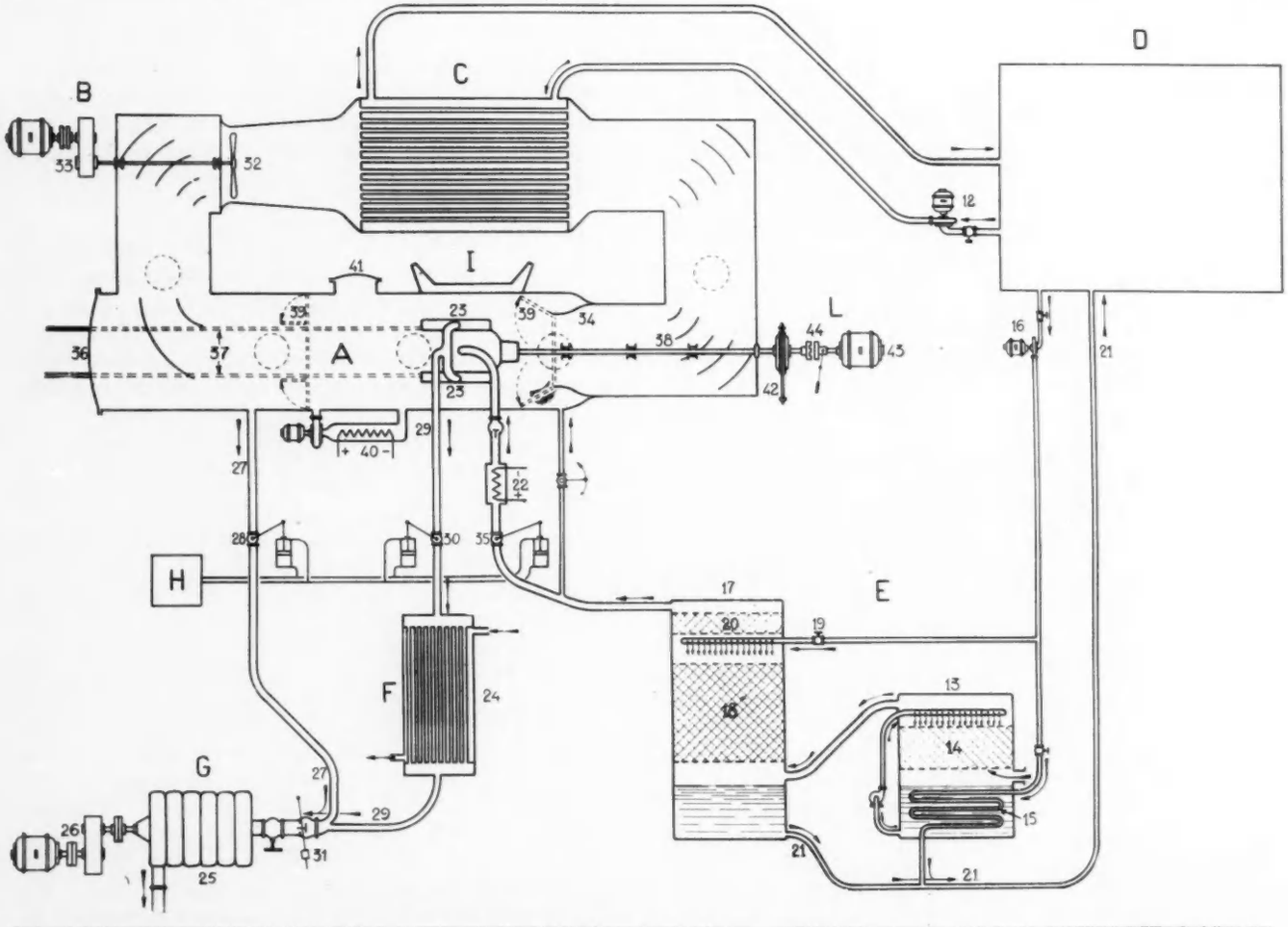


Fig. 1—General Operation Diagram

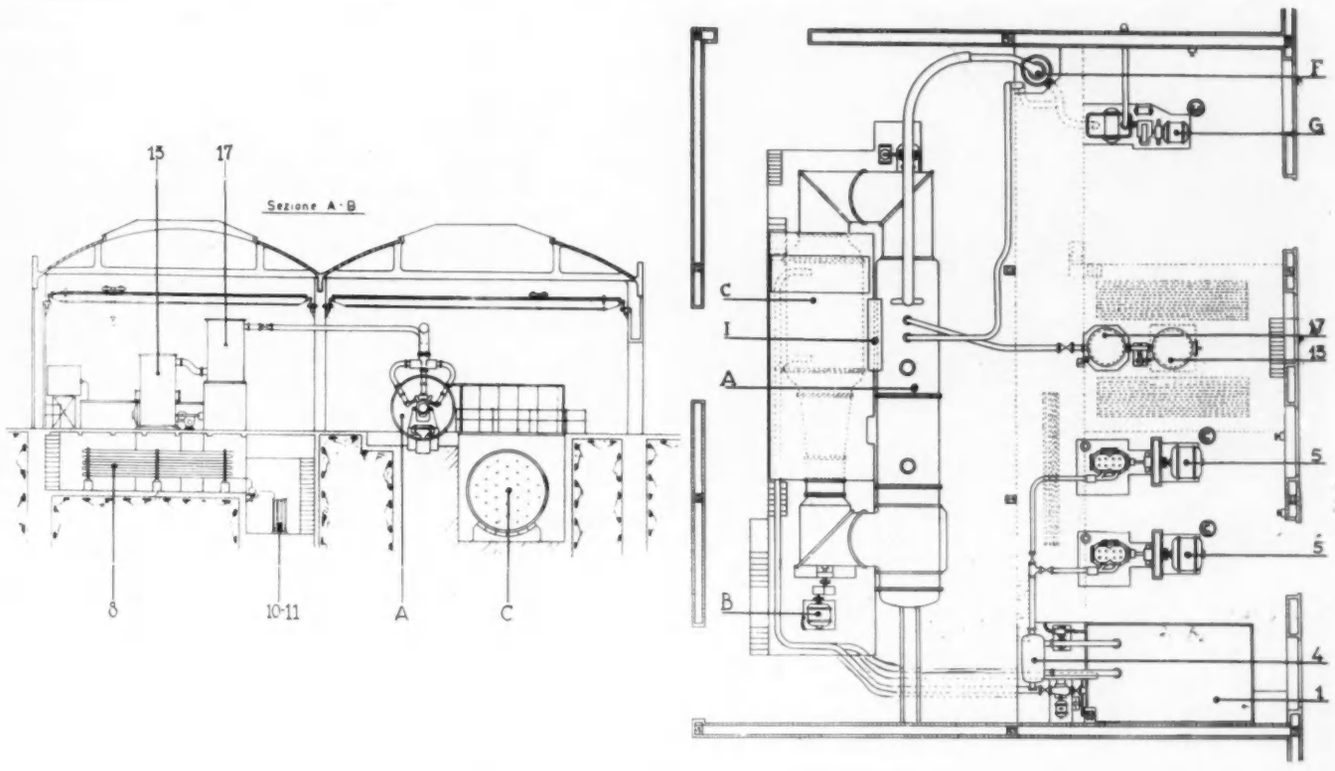


Fig. 2—General Installation Layout

conditions of temperature, weather and latitude; and then to consider what test apparatus would be necessary to simulate all the conditions which could be found anywhere at any time in this vertical column of air. In other words, the test-plant should be able, upon occasion, to duplicate quickly, conveniently, and accurately, any combination of atmospheric conditions that might be found at any point in this imaginary air-column. In fact, the evolution of modern high-altitude aircraft-engines has been made possible to a large extent by making use of such special test-plants capable of reproducing in the test-room the atmospheric conditions in which the engines must operate in flight, and it is just this that the Fiat Co. has accomplished in the construction of its new High-Altitude Aircraft-Engine Test-Plant at Torino, Italy.

The Fiat test-plant for high-altitude aircraft-engines, as it is at present, has the capacity for testing aircraft engines within the three following sets of limitations:

(1) Air-cooled engines of 500 hp. or less at constant power up to an altitude of 16,000 ft., and beyond this up to 32,000 ft. with decreasing power in proportion to the altitude

(2) Water-cooled engines of 1000 hp. or less at constant power up to an altitude of 16,000 ft. and beyond this up to 32,000 ft. with decreasing power in proportion to the altitude

(3) Water-cooled engines of 500 hp. or less at constant power up to 32,000-ft. altitude.

The capacity of the test-plant can readily be increased if future requirements should make such increase necessary. It is well known that the output of an engine when measured under random conditions must be corrected to operating conditions in "standard air", in order to compare results. Standard air contemplates a surrounding of "perfect" atmosphere characterized by a definite pressure, temperature and humidity, corresponding to the altitude. The characteristic curves of an engine tested in a perfect atmosphere should consequently be free from the effects of special and irrelevant conditions, such as calm or high winds; rain, snow or sunshine; high or low temperatures, and the like.

In a surrounding of standard air, all variations of altitude therefore resolve themselves into corresponding and contemporaneous variations of pressure, temperature and humidity. A high-altitude test-plant consequently must be able to produce these three variations and to maintain any desired atmospheric condition. Furthermore, if the engine under test happens to be cooled by air, the test-plant must also be able to supply the wind-stream that the engine would encounter during actual flight. A high-altitude test-plant must therefore, of necessity, comprise the following:

(1) An airtight test-chamber of suitable dimensions
(2) An engine test-bench
(3) A plant for reduction of the pressure of the air
(4) A refrigeration plant
(5) A humidification plant
(6) A wind-stream plant, and
(7) A central control-board.

The requirement of the airtight chamber in which the required atmospheric conditions can be maintained to conform to those under which the engine will operate in actual flight is always cause for some concern; namely, the danger of explosion. To avoid this danger in the Fiat plant, the following precautions have been taken:

(1) The provision of an automatic alarm designed to give immediate warning, should explosive mixtures find their way into the free interior of the test-chamber, and an automatic safety device for rapid evacuation of such mixtures from the test-chamber

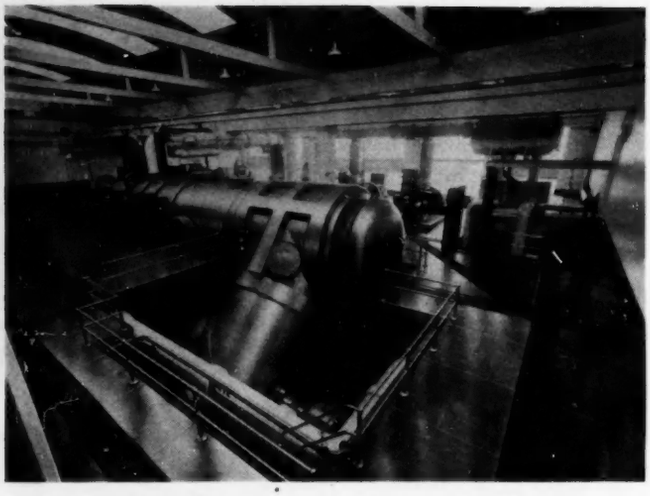


Fig. 3—The First Section, Containing the Hermetical Cell, the Test Bench, the Blower Plant and the Control-Board

(2) Complete independence of, and insulation between, the airflow, which maintains the desired atmosphere inside the test-chamber, and the intake and exhaust manifolding and relative piping in the engine test-chamber, in order that, even though constantly maintaining the required altitude-characteristics, a backfire which might occur could not possibly cause ignition of any explosive mixture that might have found its way into this standard atmosphere

(3) Refrigeration of the test-plant by means of a brine circulation-system instead of ammonia, to avoid all possibility of ammonia escaping into the test-chamber and thus causing explosive mixtures to be formed

(4) Hydraulic test of the test-chamber at a 12-atmosphere pressure; that is, one and one-half times the maximum pressure of a possible explosion, together with the installation of suitable diaphragm relief-valves set to operate at a pressure as low as 1½ atmosphere.

Another requirement of a test-plant of this kind is that it should possess a certain "thermal inertia" in order that it can maintain its regularity of operation in the case of some brief interruption or a sharp variation of the working conditions of the engine, such, for example, as a zoom or a rapid descent.

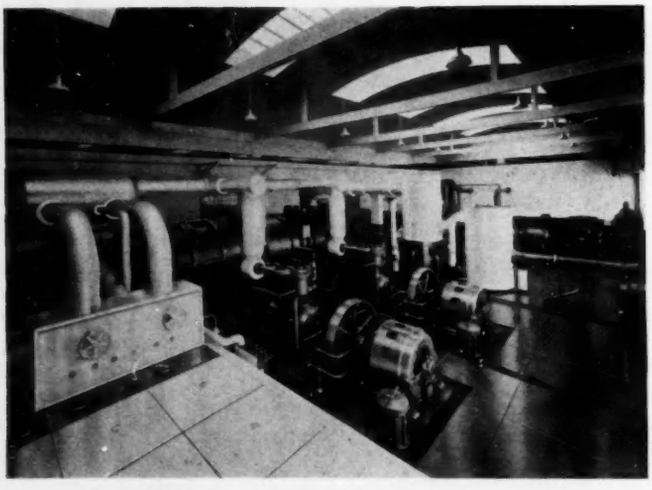


Fig. 4—The Second Section, Containing the Depression Plant and the Refrigeration Plants

At the same time, the plant should have a certain elasticity of control as well as the ability to adjust itself quickly and with precision to the conditions. Both of these requisites have been realized in the Fiat plant by means of the following arrangements:

(1) Refrigeration of the test-plant by means of brine circulation capable of maintaining the rate of heat absorption for long periods, instead, as already mentioned, of direct circulation of ammonia

(2) The installation of a system of two independent controls, a thing made possible by the aforementioned method of cooling by brine circulation; that is, an immediate direct one operating on the brine circulating-system, and a much slower indirect one operating on the ammonia-compressor speed

(3) Installation in the various air conduits, and in the engine oil and water-circulation systems, of heating coils capable of rapidly altering the operating conditions of their respective systems, independently from the general operating conditions of the test-plant.

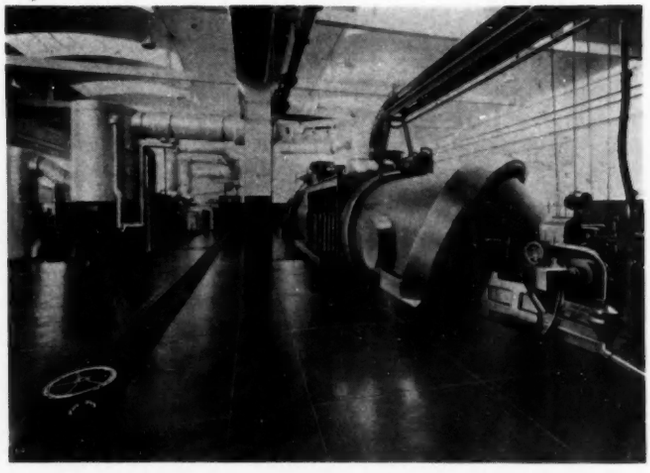


Fig. 5—Rear View of the First and the Second Sections

A last but important requisite in a test-plant of the kind under consideration consists in the rational layout of the various control systems, so as to offer the possibility of keeping the whole test-plant "under the eye" and "under the hand" of the observer at the control-stand. In the Fiat plant, this has been attained as follows:

(1) All controls and indicators have been brought together into a single "central control-board", several unusual and special devices having been adopted to attain this

(2) The operation of all controls for decreasing pressure in proportion to altitude is obtained simultaneously and automatically at all points by means of a special apparatus.

The operation of the test-plant is shown diagrammatically in Fig. 1, for more simple presentation, and may be divided into the following principal units: (*a*) the refrigeration system, (*b*) the engine fuel-feed system, (*c*) the engine lubrication-system, and (*d*) the engine water-cooling system.

The installation of the test-plant, shown in Fig. 2, occupies an area approximately 70 x 85 ft., the roof being about 32 ft. high. This space is divided into two sections, both being served by a 4-ton electric-crane. The first section contains principally the test-chamber, the wind-stream plant, and the central control-board. The second section contains the air-pressure-reduction plant, and the refrigeration plant.

Fig. 3 shows the first section, containing the hermetical

cell, the test bench, the blower plant and the control-board. The second section, shown in Fig. 4, contains the depression plant and the refrigeration plants. A rear view of the first and the second sections is shown in Fig. 5.

The Engine Test-Cell

The test-cell, Figs. 6 and 7, consists of a huge circular-shaped tube which forms the upper main section of a link-shaped, rectangular ring of similar cross-section, resting in an almost vertical position on its lower main section. The upper main section, that is, the engine test-cell itself, is on the level of the test-floor, while the lower main section, upon which the entire ring rests, is sunk in a trench below floor level. The total axial developed length of the ring is about 131 ft., the overall assembled length is approximately 62½ ft., the diameter of the section in the upper main branch is nearly 9 ft., in the lower main branch it is 5½ ft. in one place and about 11 ft. in another. The total volume of the enclosed space is approximately 6000 cu. ft.

The shell of the test-cell, as well as all other parts of the ring, is built of steel plates 1⅛ in. thick, with welded joints and with wide connecting flanges and heavy strengthening ribs, all completely covered with a layer of heat-insulating expanded-cork, ranging in thickness from 1⅛ to 1½ in. This insulation is furthermore covered with a compound of tar to eliminate all possibility of air seepage and the consequent condensation of water between the insulation and the metal surfaces of the test-chamber.

The inclined position of the test-chamber is intended not only to reduce the floor space required, but also to avoid circulation of enclosed air, with the consequent losses which would occur during periods when the chamber would necessarily be left open, if the cooling surfaces were placed on the same level as the engine test-cell. The round section of the chamber, the accurate welding of all inside joints of the shell, the careful sand-blasting and zinc-plating of all inside walls, and the installation of well-designed airstream-deflectors in each of the four huge elbows of the chamber, all contribute to the efficient production of a high-velocity artificial-wind within the chamber. The method of gasketing the open end of the chamber insures a remarkable degree of airtightness. The thickness of the layer of insulation was calculated so that whatever losses might occur due to heat transmission through the walls of the chamber would be compensated for many times over by the capacity of the refrigerating plant. That ample control in this respect has been obtained is shown by the fact that an operating temperature of minus 58 deg. fahr. can be maintained in the chamber.

The whole weight of this huge link-shaped chamber is supported on saddles, and its axial motion is controlled by means of foundation bolts through the saddles supporting the rear ends only of each main branch of the chamber; thus,

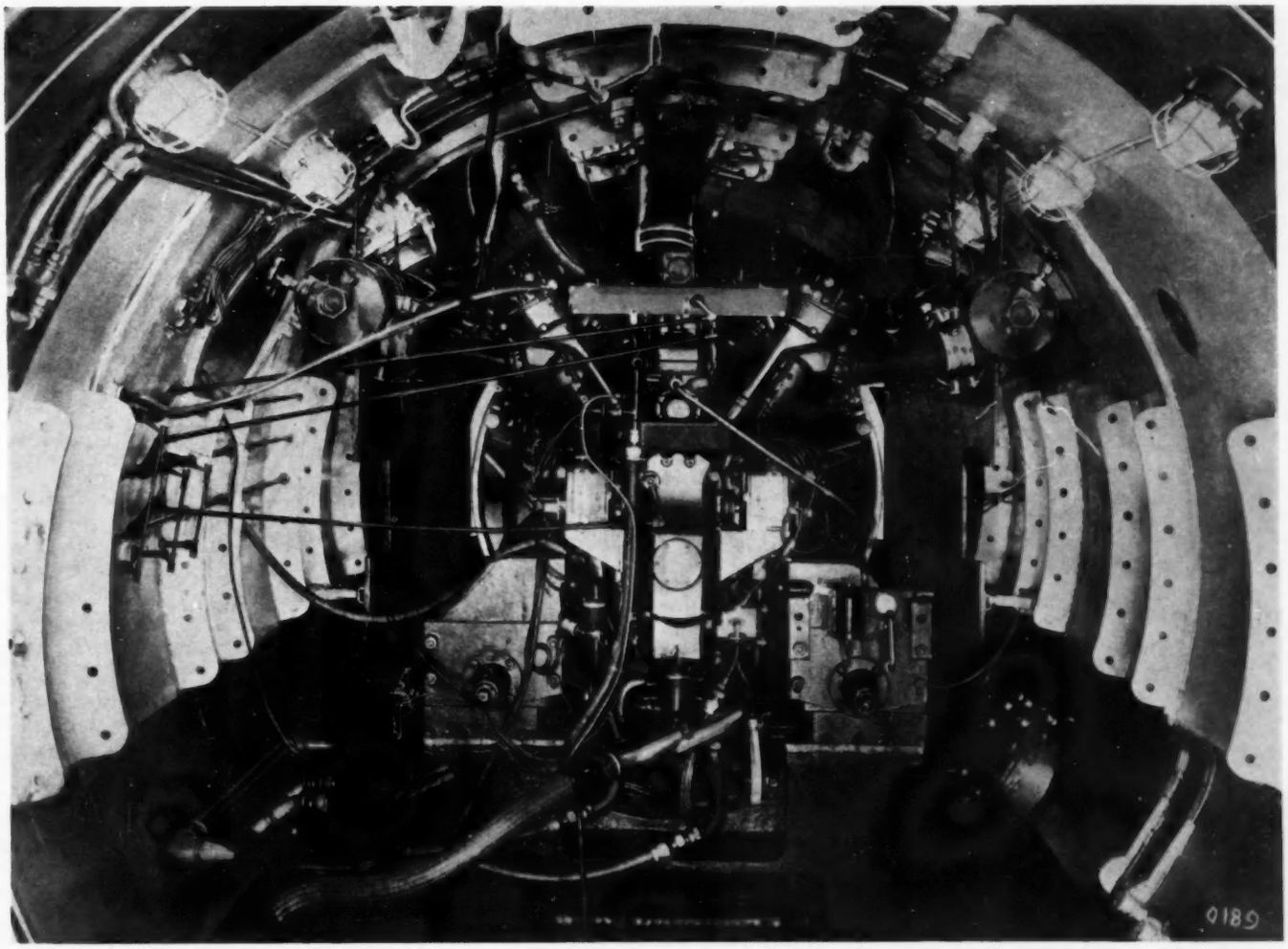


Fig. 6—The Inside of the Engine Test-Cell

expansion and contraction due to variations of temperature can freely take place without causing distortion or overstressing of any members of the structure, which might result in leaks into the shell.

The test-chamber contains three principal groups of apparatus corresponding to the three functions which they perform:

(1) The engine test-cell, which is in the main upper branch of the chamber

(2) The wind-stream plant, which is in the smaller portion of the lower main branch

(3) The air-cooling surfaces, which are in the larger portion of the lower main branch

(4) Then, of course, there are the two inclined end-sections of the chamber each with two series of airstream deflectors at the elbows. Otherwise these two sections are empty, acting only as air passages between the upper and the lower main branches of the chamber.

The Refrigerating Plant

The refrigerating plant, a diagram of which is shown in Fig. 8, comprises:

(*a*) A refrigerating machine; located at *D* in Fig. 1

(*b*) The cooling surfaces which absorb heat from the air enclosed in the test-chamber; located at *C* in Fig. 1

(*c*) The cooling surfaces which absorb heat from the air fed to the carburetor of the engine; located at *E* in Fig. 1

(*d*) The cooling surfaces which absorb heat from the exhaust gases flowing from the engine; located at *F* in Fig. 1.

The refrigerating machine is of the ammonia-absorption type, using a circulation of a calcium-chloride-brine solution of 32 deg. Baumé density and a freezing point of minus 67 deg. fahr., which, when the test-plant is in operation, is kept constantly circulating over the aforementioned cooling surfaces.

The refrigerating machine comprises, as shown in Fig. 8:

(*a*) The brine tank (1), with the dimension of 25 x 14½ x 8 ft., in which the brine is cooled by the evaporation of the ammonia in the submerged banks of steel cooling-tubes (2). Two motor-operated vane-type agitators (3) keep the brine in circulation over the cooling-tubes

(*b*) An ammonia-vapor separator (4), placed above the brine tank, into which flows any ammonia vapor formed inside the cooling-tubes

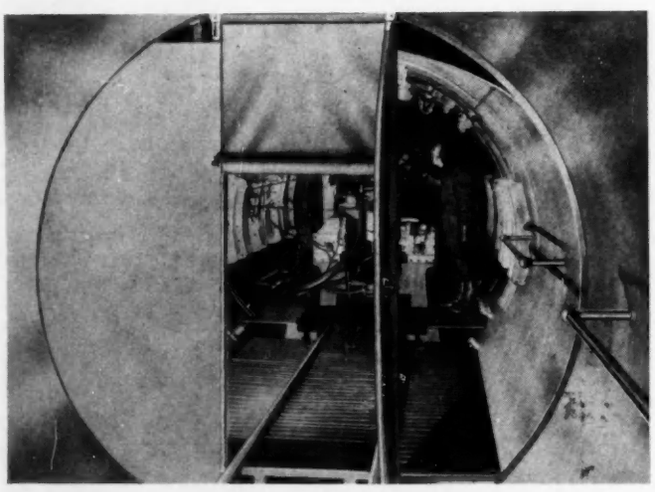


Fig. 7—Inspecting the Engine, After Having Shut Down and Warmed the Test-Cell

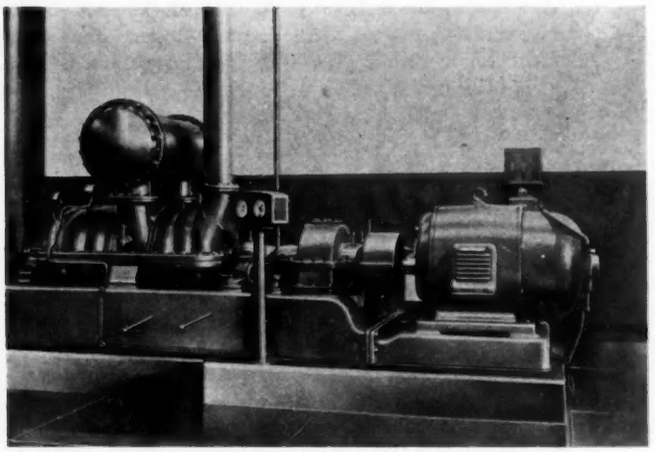


Fig. 9—Turbo-Type Blower-Plant for Depression Inside the Test-Cell and at the Induction and Exhaust Ports of the Engine

(*c*) Two motor-compressors (5), for the purpose of exhausting any such vapor, and which are connected in multiple in the ammonia-vapor line. These compressors are of the two-stage reciprocating-type, each driven at 250 r.p.m. by a 240 hp. three-phase induction-motor. Liquid ammonia is injected between the high and the low-compression cylinders to absorb the excessive heat in the compressed vapor after the first stage. The compressed saturated ammonia-vapor is then led into the oil separators

(*d*) Two oil separators (7), one in series with each compressor, for cleaning the ammonia vapor of any oil that may have been carried over from the compressors

(*e*) Two condensers (8), of usual construction, one in series with each separator, which receive the ammonia vapor from the oil separators and condense it into liquid ammonia at about 68 deg. fahr.

(*f*) Supplementary cooling coils (10) (11), in which the liquid ammonia from the two condensers is further cooled to about 32 deg. fahr. The first stage of coils (10) is cooled by a current of fresh water, while the second stage (11) is cooled by the passage of liquid ammonia over the coils. The cold, liquid ammonia thus obtained is finally led to the ammonia-vapor separator

(*g*) An ammonia-vapor separator (4), from which any free vapor is exhausted into the twin condensing-system by the two aforementioned motor-driven compressors (5), there to again take up its cycle as just outlined.

The cooling-surfaces, *C*, in the test-chamber, consist of a bank of 2526 seamless steel-tubes located in the larger portion of the lower main branch of the test-chamber. These tubes have a total effective surface area of about 18,800 sq. ft. Around these tubes there is a counter-flow of brine from the main brine-tank (1), through suitable piping in which are placed the necessary control valves and a delivery pump (12) It is obvious that an adjustment of the brine-flow control-valve will effect a corresponding change of the temperature of the air flowing to the engine test-cell inside the test-chamber.

The cooling-surfaces, *E*, see also Fig. 1, in the air-feed flow to the engine, are contained in two units in series, each of riveted sheet-steel construction. Both units are completely covered with insulating material which, in the case of the first unit, is ⅜ in. thick and, in the case of the second unit, is 1 in. thick.

(*a*) The first cooling-unit (13) is 10 ft. tall and 5 ft. in

diameter, and contains a thick filter (14) consisting of a mass of tiny metal-rings which provide a large active cooling-surface. Water flowing in a closed circuit runs continuously in small streams onto this filter. A centrifugal pump raises the water to the jets above the filter, and a counter-flow brine-cooling coil keeps it at the desired temperature. This brine comes directly from the main brine-tank through piping in which are installed a control valve and a pump (16). Air for the engine is taken either from the test-room or from out-of-doors by the normal suction of the engine under test, and, while passing through the Raschig filter in contact with the descending cooling water, undergoes a reduction of temperature down to about 40 deg. fahr.

(*b*) The second cooling-unit (17) is 8½ ft. high and 5 ft. in diameter, and contains a second filter (18). The cooling in this second unit is by means of a continual shower of brine which is by-passed from the same brine-delivery piping that cools the water in the closed circuit of the first cooling-unit. This second unit has its own control (19), and is capable of further reducing the temperature of the air down to about minus 30 deg. fahr. A third filter (20) is placed above the brine shower, so that the air flowing to the engine must pass through it. The object of this third filter is to free the air fed to the engine of all moisture in the form of drops or spray which might conceivably pass along with the moving air. It must be noted, however, that the rate of flow of the air through this second cooling-unit has been kept so low that all liquid in suspension in the air would separate itself by gravity before passing on to the engine, even without the presence of the third filter.

The free brine settling to the bottom of the second unit, together with that which has passed through the water-cooling coil of the first unit, finds its way back to the main brine-tank through the same system of piping. It is obvious that controlling the flow of brine to each of these units allows a wide range of temperatures to be obtained in the air fed to the engine.

The cooling-surfaces, *F*, as in Fig. 1, placed in the flow of the exhaust gases from the engine, comprise an independent group, consisting principally of:

(*a*) Two sheet-metal exhaust-gas collectors (23), placed in the test-cell just above and on each side of the engine, and surrounded by sheet-metal water-jackets. The exhaust gases from the engine, at a temperature of about 1475 deg. fahr., undergo here a first cooling

(*b*) The exhaust-gas cooler (24), which is about 28½ ft. high by 4½ ft. in diameter, contains a bank of 180 seamless-steel cooling-tubes over which passes the same water that is used for the ammonia condensers—(8), as in Fig. 8—already described. This water enters the cooler at a temperature of 72 deg. fahr. and leaves it at a temperature of about 86 deg. fahr. In this way the exhaust gases passing through the cooler can be brought down to a temperature of about 140 deg. fahr.

This type of cooler was adopted to avoid the development of a considerable amount of water vapor in the gases, which would cause corrosion in, and greatly overload, the exhaust turbo-blower *G*, as in Fig. 1. The exhaust-gas cooler was tested under a pressure of about 110 lb. per sq. in. to guard against damage from possible explosions in the engine exhaust-line.

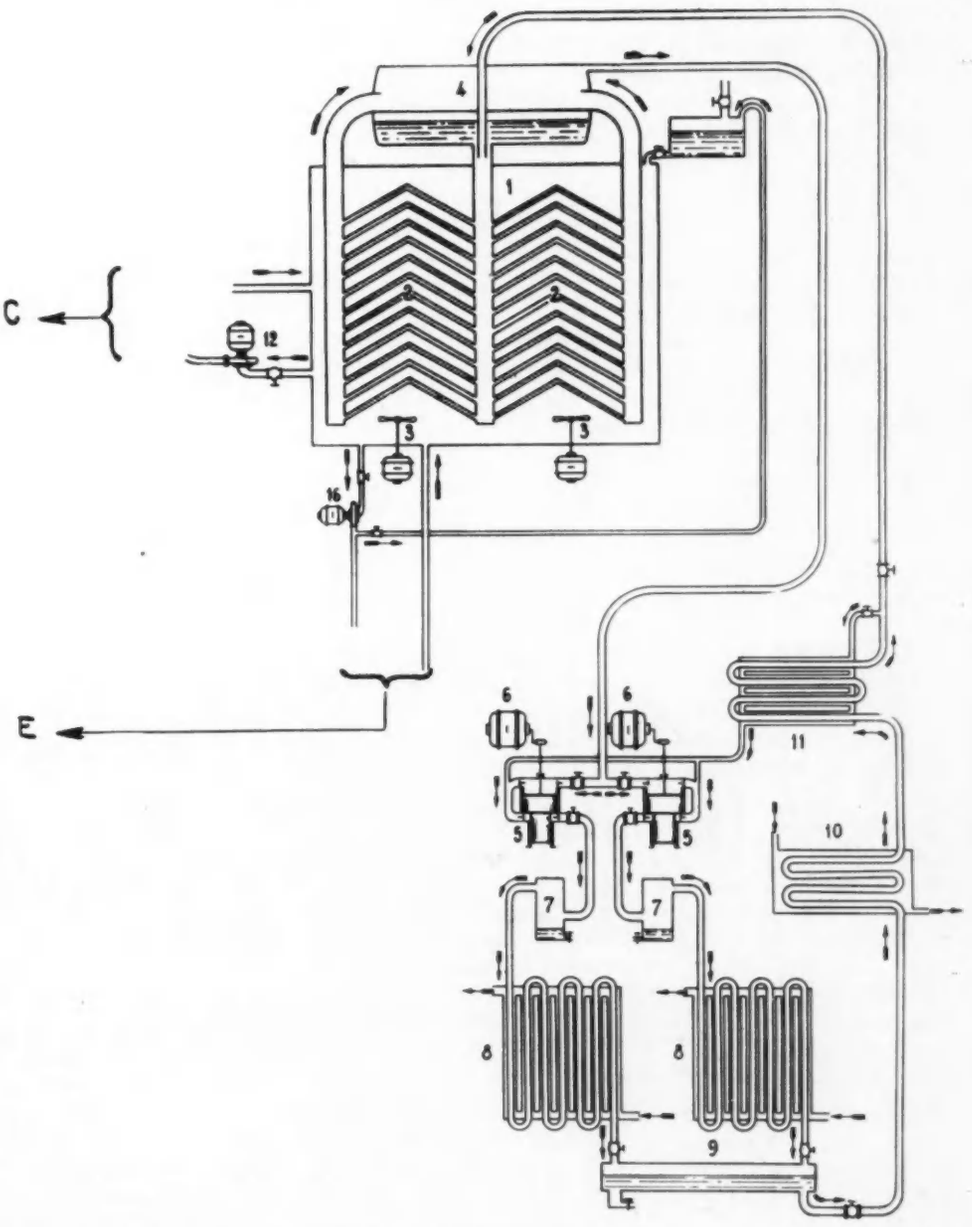


Fig. 8—The Main Refrigerant Plant

The entire refrigerating plant is at present capable of normal operation down to minus 30 deg. fahr. with an output of about 133,000 B.t.u. per hr. The plant is, however, so designed as to maintain temperatures of about minus 60 deg. fahr. with an output of approximately 250,000 B.t.u. per hr.

All piping is covered with the same type of insulation as that adopted for the test-chamber and the cooling-units, the thicknesses varying from ½ in. to 1⅛ in. Furthermore, all piping was tested at considerably above the normal working pressures.

The Air-Pressure-Reduction Plant

The air-pressure-reduction plant includes principally:

(*a*) A seven-stage turbo-blower, *G*, as in Fig. 1, and shown in Fig. 9, run at 750 r.p.m. by a three-phase induction-motor (26) of 300 hp. through a gear-train which steps up the velocity of the blower to 9000 r.p.m.

(*b*) The test-chamber pressure-reduction piping leading to the turbo-blower in which there is an automatically operated control-valve (28), which in turn is controlled by an automatic altitude-regulator, *H*, as in Fig. 1. This arrangement makes it possible to obtain pressures in the test-chamber as low as a value of 0.26 for the ratio of the absolute pressure obtained divided by the absolute normal atmospheric pressure, which corresponds to an altitude of approximately 32,700 ft. A diagram of the altitude-regulator is shown in Fig. 10.

(*c*) The exhaust-gas-cooler piping (29) leading from the exhaust-gas cooler to the turbo-blower, in which there is also an automatically operated pressure-control valve (30), operated by the same type of automatic altitude-regulator as that used in the test-chamber reduction-piping, thus making it possible to maintain the same pressure in the engine exhaust-piping as that maintained in the test-chamber.

These two sets of piping are connected to the turbo-blower through a single manifold in which a check-valve is placed to protect the blower in case of a sharp increase of pressure, such as would occur for example in the case of an explosion inside the test-chamber. The blower discharges outside the test-room.

The Artificial-Wind Plant

The artificial-wind plant, *B*, as in Fig. 1 and shown in Fig. 11, comprises principally:

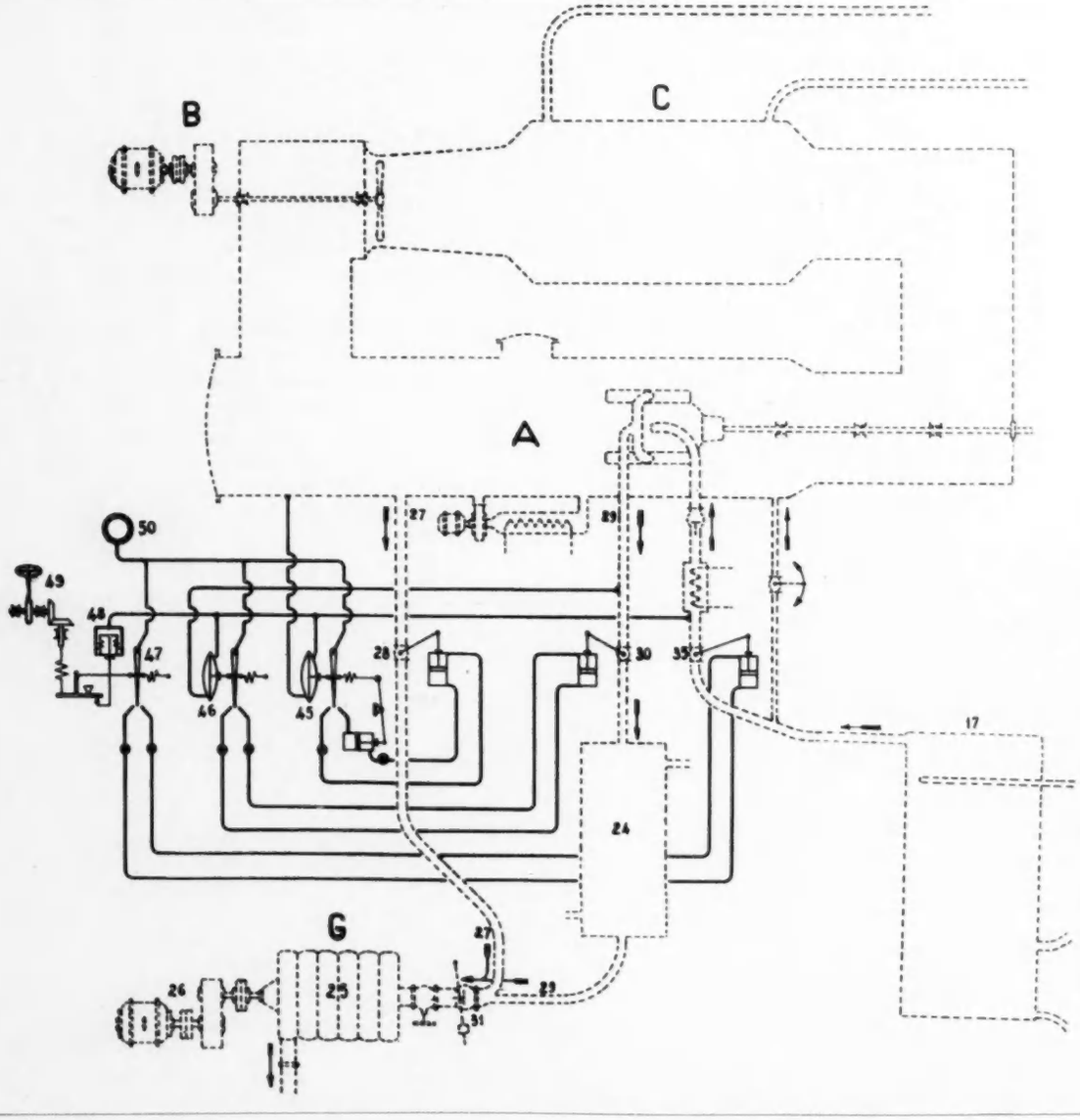


Fig. 10—Diagram of the Automatic Altitude - Regulator

(45) Regulator of the Depression in the Ambient Air Inside the Hermetical Cell
(46) Regulator of the Depression in the Exhaust - Gas Flow from the Engine
(47) Regulator of the Depression in the Fuel - Feed Flow to the Engine
(48) Pressure Valve of the Regulator Plant
(49) Hand-wheel Register of the Regulator Plant
(50) Oil-Pump of the Regulator Plant

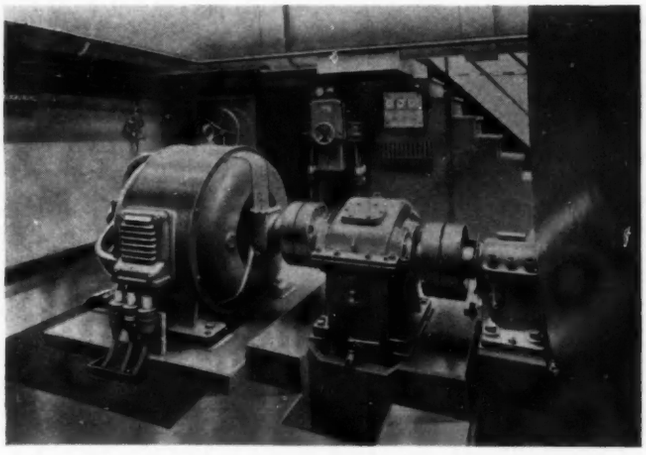


Fig. 11—Turbo-Type Blower-Plant for Providing the Wind-Stream Inside the Test-Cell

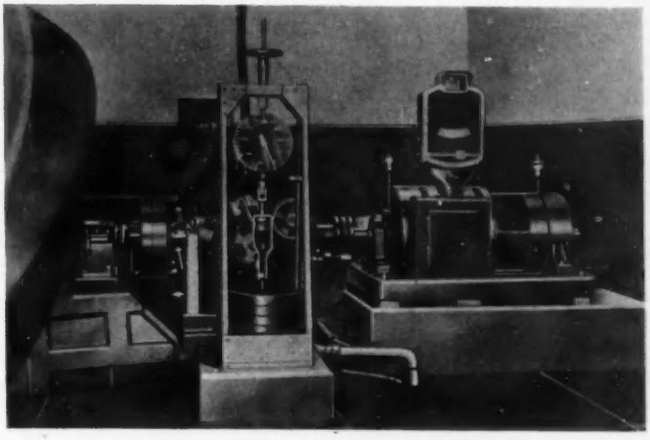


Fig. 12—The Power-Absorption Brake

(*a*) A propeller-type fan (32), mounted in the narrower portion of the lower main branch of the test-chamber

(*b*) A driving motor (33) which drives the fan and is mounted outside the test-chamber below the floor level. This is a 300 hp. three-phase induction-motor running at 975 r.p.m. and it drives the fan at 2400 r.p.m. through a suitable step-up gear-train.

The wind thus created, moving in a closed circuit within the test-chamber, can be made to reach a velocity of 100 m.p.h., which corresponds to the movement of 2650 cu. ft. of air per sec., at a pressure corresponding to an altitude of 16,300 ft. The total drop in pressure which occurs in the circuit under these conditions is 6½ in. of water, and is localized as follows:

3¾ in. in passing over the cooling-surfaces, *C*

1¼ in. in the engine test-cell

1½ in. in the remaining parts of the test-chamber.

When testing an air-cooled engine, the walls of the chamber just at the entrance of the engine test-cell (34) can be so altered in shape as to direct the wind-stream directly upon the engine.

The Humidifying Apparatus

The apparatus for humidifying the engine intake-air (35) is placed between the second air-cooling unit (17) and the engine, as in Fig. 1. Naturally, this air always leaves the second cooling-unit in that state of water-vapor saturation corresponding to its temperature and to the outside atmospheric pressure. The humidifying apparatus includes:

(*a*) A pressure-reducing valve (35) operated by the same automatic altitude-regulator, *H*, that has already been mentioned. This valve automatically reduces the pressure of the intake air to that corresponding to the altitude of the test in progress

(*b*) A heating chamber (22) fitted with electric heating-coils operated by a rheostat located on the central control-board, *I*. The air is warmed in this chamber up to the temperature corresponding to the altitude of the test under way, and is then led to the air intake on the engine.

It is clear that this operation of altering the pressure and temperature of the intake air causes a corresponding variation in its humidity. This is shown by an hygrometer mounted on the central control-board, *I*. It is furthermore evident that if the intake air is made to come from the cooling-units at a temperature sufficiently low, it will be possible by the devices just mentioned to effect an almost immediate and simultaneous change of its pressure, of its temperature, and, within certain limits, of its humidity.

The Engine-Test Apparatus

The upper main branch of the test-chamber can be divided into three sections, as shown in Fig. 1; namely:

(*a*) A front section, provided with a removable dished head (36) and an industrial track (37) for placing the engine to be tested in position on the test-block within the test-cell

(*b*) An intermediate section, *A*, forming the engine test-cell

(*c*) A rear section, through which the shaft (38) passes which connects the engine under test in the cell with the power-absorption brake, *L*, outside the test-chamber, as shown in Figs. 12 and 13.

The engine test-cell itself is provided with:

(*a*) A port-hole fitted with a triple-thick safety-glass through which it is possible to observe the operation of the engine from the outside

(*b*) A manhole (41), through which it is possible to enter the cell for close inspection of the engine under test

(*c*) Two thin insulating bulkheads (39), hinged and detachable, by which it is possible, when there is little or no wind-stream, to close and insulate the engine test-cell from the rest of the test-chamber

(*d*) A heating apparatus (40), consisting of an electric fan and an adjustable electric heating-unit by means of which it is quickly possible to warm up the engine test-cell, after having closed it with the bulkheads already mentioned, so as to obtain therein satisfactory working conditions for the test crew.

The shaft (38), between the engine under test and the power-absorption brake, is fitted with flexible couplings and three adjustable bearings. To avoid troubles due to torsional vibrations, the test plant is provided with two readily interchangeable shafts, which have critical speeds which are quite different, the one from the other.

The entire power-absorption brake-unit is mounted on a bracket, which is bolted to the outside of the test-chamber so that expansions and contractions of this latter will not affect shaft alignment. This unit consists of two pieces of apparatus, namely:

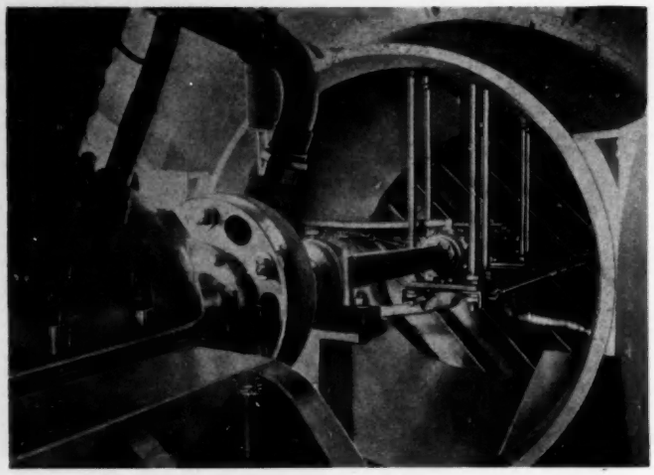


Fig. 13—Connecting Shaft between the Engine Inside the Test-Cell and the Power-Absorption Brake Outside the Test-Cell

(*a*) An hydraulic power-brake (42), of the Froude type, directly connected to the shaft already mentioned

(*b*) An electric-dynamometer brake (43) that can be coupled to the free end of the Froude brake-shaft by means of a sliding toothed-coupling (44). By using this electric dynamometer as a motor, it can be made to serve either as a starter or as a means for measuring mechanical losses of the engine under test.

The power-absorption brake-unit, *L*, is operated from the central control-board, *I*, through relays to the various control points on the equipment. An interesting device, shown in Fig. 14, has been adopted for the reading of the dynamometer dial; the pointer of the dial is fitted with a small electric lamp so that its image is reflected through the necessary prisms onto a correspondingly graduated opaque-glass dial at the central control-board.

In addition to the power-measurement apparatus mentioned, there is of course also to be found: A fuel-feed system, shown in Fig. 15; a lubrication system, shown in Fig. 16; and a cooling-water supply-system, shown in Fig. 17.

The fuel-feed system, Fig. 15, consists mainly of:

(*a*) A fuel-mixing tank, which consists of a metal container to which is connected piping leading from six different tanks, each containing a different kind of fuel. By this means any desired mixture can be obtained easily and also variations of fuel can be made while tests are in progress

(*b*) A gauze filter inserted in the fuel line after the mixing tank already mentioned

(*c*) A first float-chamber fitted with a breather, and therefore operating at atmospheric pressure

(*d*) Four fuel-meters inserted in the fuel piping in multiple, two being of the volume type with capacities of 5 litres and 2½ litres, respectively, and two being of the direct-reading type, thus permitting immediate and continuous control of fuel consumption during the test. The four fuel-meters are all mounted on the central control-board

(*e*) A second float-chamber which has a pressure-equaliz-

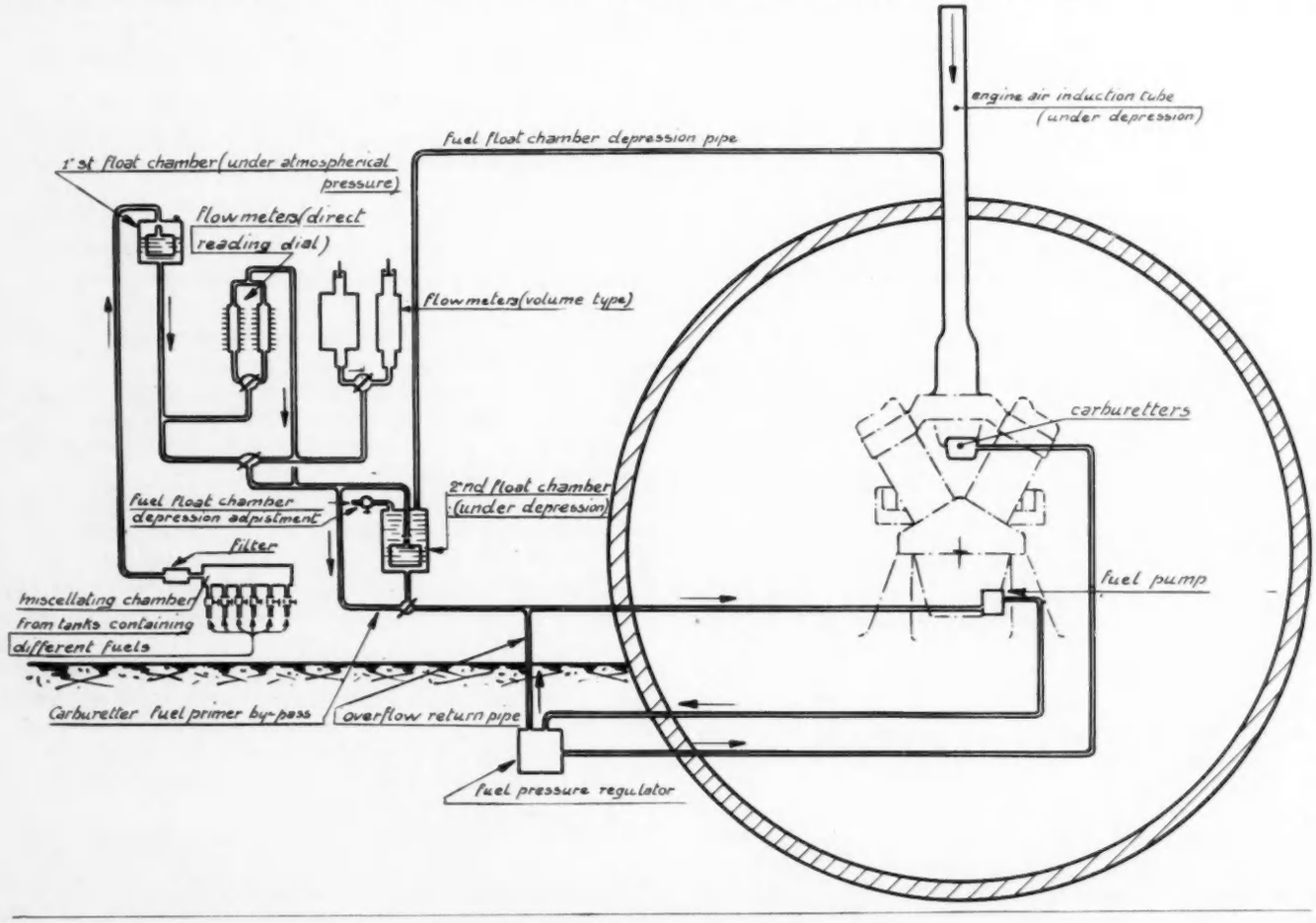

engine air induction tube (under depression)
fuel float chamber depression pipe
1st float chamber (under atmospherical pressure)
flow meters (direct reading dial)
flow meters (volume type)
carburetters
2nd float chamber (under depression)
fuel float chamber depression adjustment
filter
fuel pump
miscellating chamber from tanks containing different fuels
carburetter fuel primer by-pass
overflow return pipe
fuel pressure regulator

Fig. 15—Engine Fuel-Feed System

ing connection with the engine intake-air-feed piping, and which therefore causes it to operate at the existing intake-air pressure. This float chamber is also provided with a vacuum meter mounted on the central control-board, and with a vacuum adjustment-valve for direct admission of outside air

(*f*) A fuel pump mounted on the engine under test

(*g*) A pressure regulator inserted in multiple in the fuel-feed piping and provided with an overflow return pipe to the regulator.

It should be noted that the very small quantity of fuel vapor which passes through the pressure-equalizing pipe connecting the second float-chamber with the engine intake-air-feed duct, does not affect the reading of the meters, inasmuch as these meters are located between this float-chamber and the fuel-supply tank.

The lubrication system, Fig. 16, consists of the following:

(*a*) An oil tank connected with the test-chamber by a pressure-equalizing pipe, so that it functions at the same low pressure existing therein, and provided with a vacuum gage located on the central control-board, with a vacuum adjustment-valve which admits air to the tank directly from the outside, and which is provided also with an electric heating-unit for heating up the oil preparatory to starting the engine. The entire oil tank is mounted on a weighing machine having its dial located also on the central control-board. Thus, it is possible to check at any moment the oil consumption of the engine under test

(*b*) An oil-cooling coil located in the return line to the oil tank, and consisting of a coil immersed in a counter-flowing current of fresh water

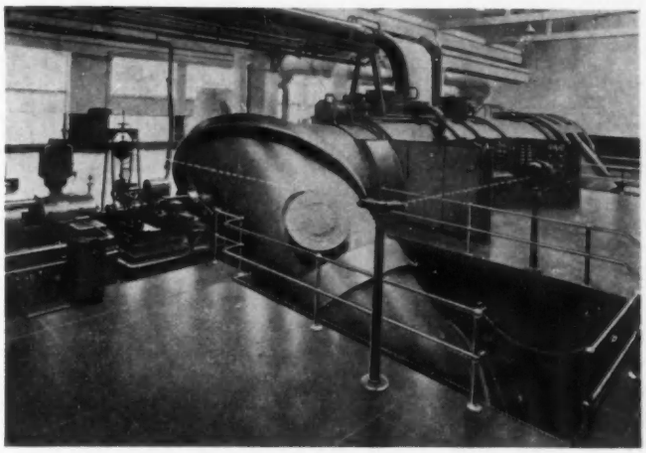


Fig. 14—Optical Relay for Reading the Power-Absorption-Brake Dial

(*c*) Two thermometers mounted on the central control-board and showing the temperatures of the oil before entering and after leaving the engine.

The cooling-water system necessary for testing water-cooled engines, Fig. 17, is made up of the following units:

(*a*) A cooling-water tank with a breather, thus allowing operation at atmospheric pressure, and provided with the usual-type water-level glass, and an electric heating-coil for heating up the water for starting the engine

(*b*) A motor-driven pump for circulating the heated water

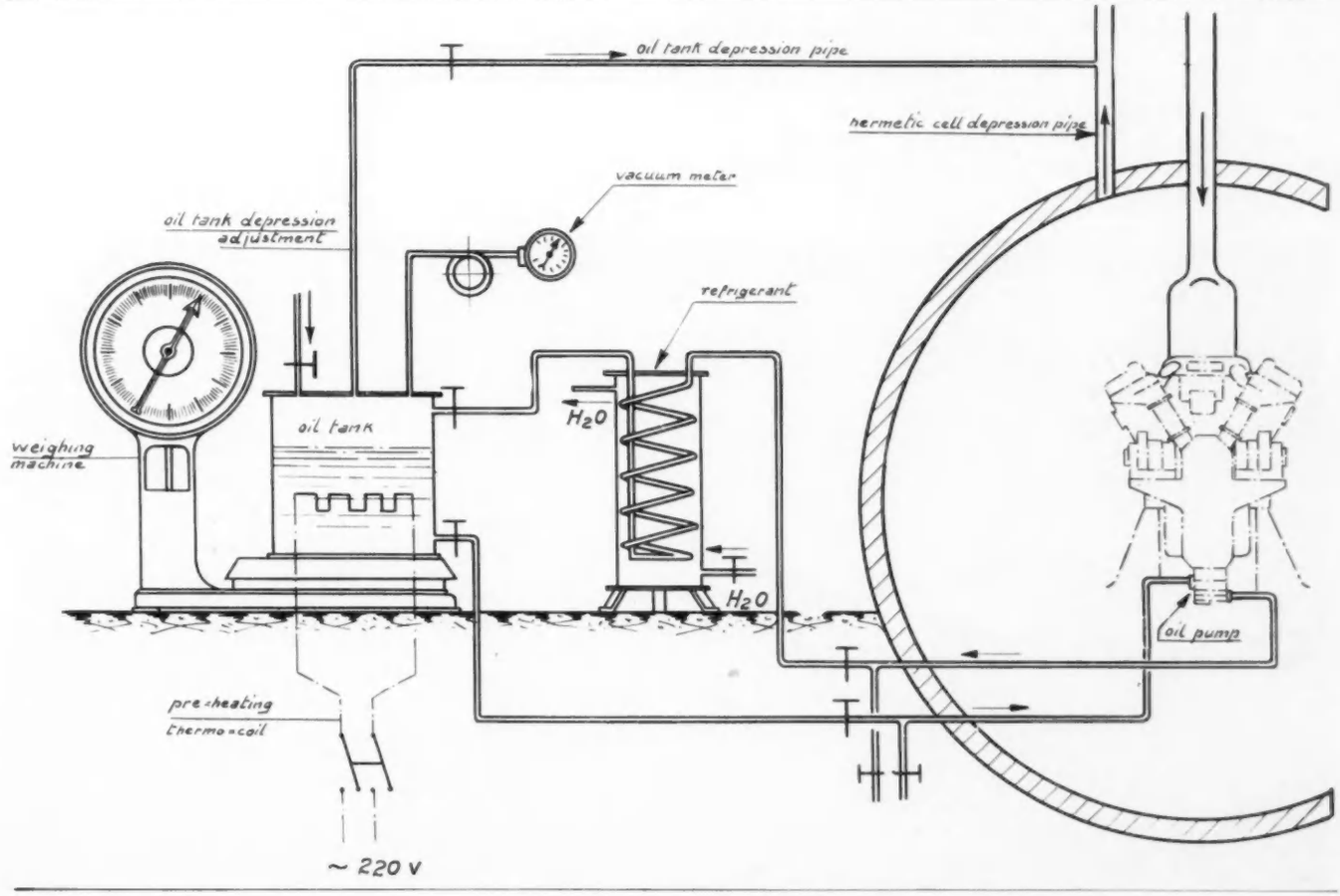


Fig. 16—Engine Lubrication-System

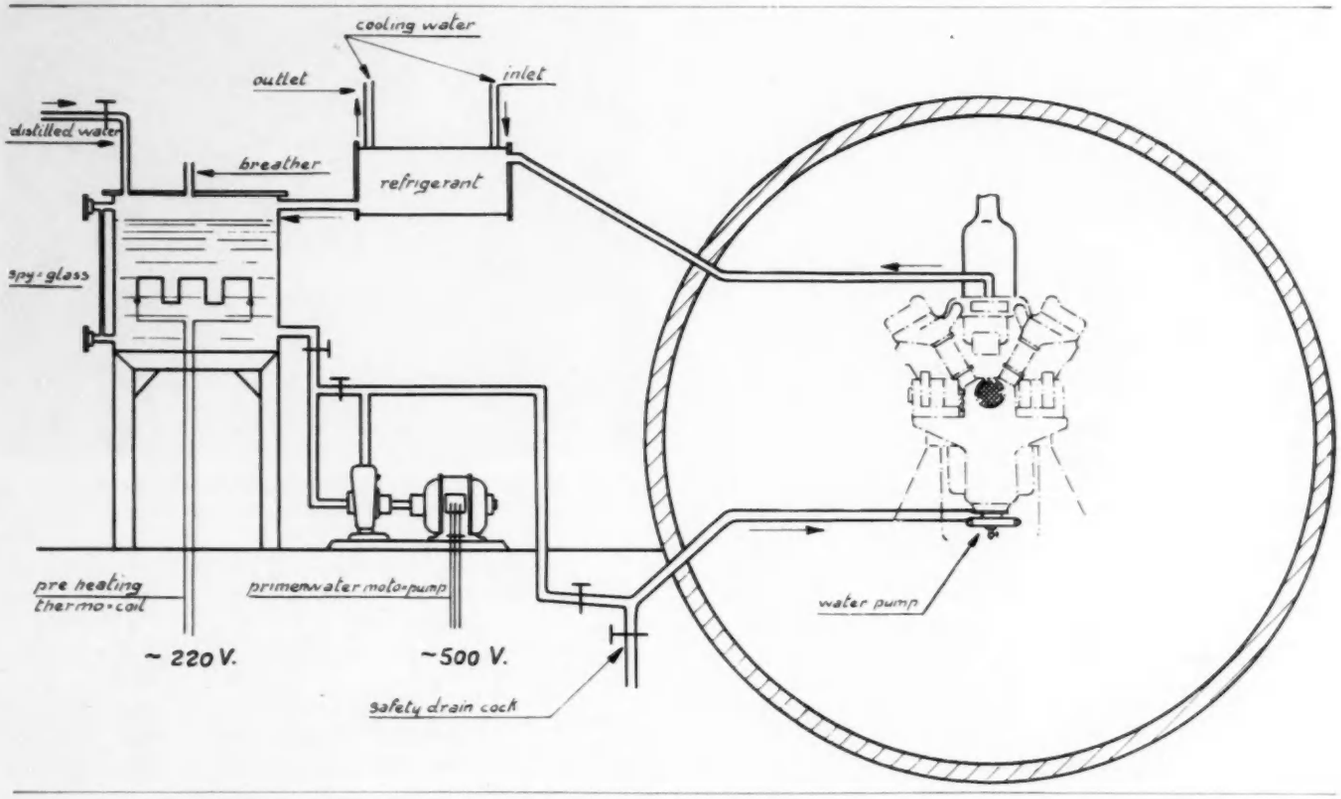


Fig. 17—Engine Water-Cooling System

preparatory to starting the engine. After the engine is started, this pump is cut out and the circulation of water is handled by the water-circulation pump on the engine

(*c*) A water-cooling coil located in the return line from the engine, and consisting of a coil submerged in a counter-current of fresh water. Thermometers registering water temperatures at the engine inlet and outlet are mounted on the central control-board.

The Central Control-Board

The central control-board, shown in Figs. 18 and 19, is installed on the test-floor, with its back against that part of the upper main branch of the test-chamber which constitutes the engine test-cell. It consists of three panels, placed in a semi-circular position for easy reading, on which are mounted all the necessary instruments and controls. Those which are

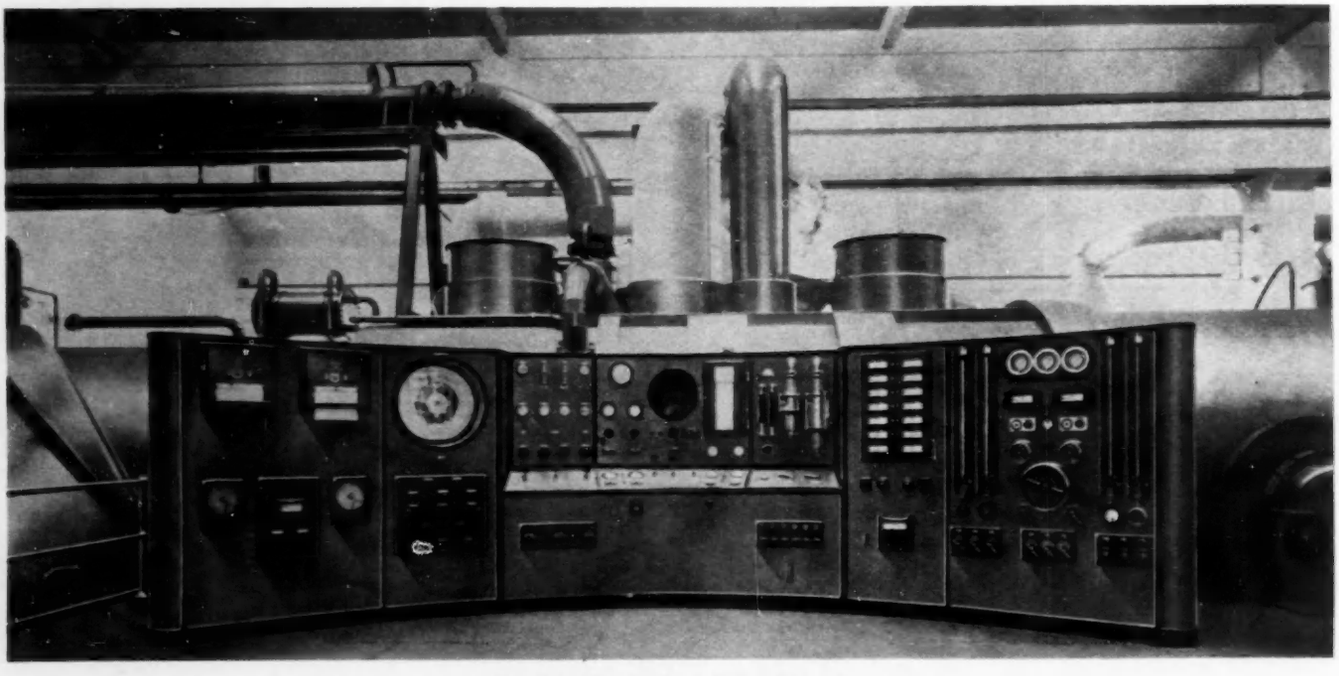


Fig. 18—The Central Control-Board

necessary for operating each single unit of the plant are grouped together, insofar as this is possible. The control board was laid out so that the operator always has the entire test-plant under full observation and control.

In general, the control-board is the focal point to which is brought, by means of instruments or by direct observation, all information essential to the conduct of the tests; and from which emanate, through the various controls, all the necessary variations and adjustments in the operation of the apparatus. The principal equipment includes:

(*a*) The port-hole into the test-cell, for direct observation of the engine during test

(*b*) The engine controls; namely, those for starting, spark advance, throttle adjustment, altitude, fuel mixture, and the like

(*c*) The electric tachometer, for indicating engine-speed

(*d*) The thermometers, pressure gages, and level indicators, for the fuel, oil, water and exhaust gases

(*e*) The temperature indicators and recorders, the altitude meters and barometers, for indicating the condition of the air inside the test-cell and of the intake-air flow to the engine

(*f*) The hygrometer in the engine intake-air duct

(*g*) The analyzer, for the engine exhaust-gases

(*h*) The flow meters, consumption meters, and automatic weighing machines, for the fuel, lubricant, and intake air

(*i*) The automatic alarm, which gives immediate warning in case of formation of explosive mixtures inside the test-chamber

(*j*) The hand-wheel control of the automatic altitude-regulator, and

(*k*) The "rapid-descent" control.

Most of these instruments and controls are well known, or have been already referred to. However, special mention should be made of the automatic altitude-control; see Fig. 10. This depends for its operation upon the well-known diaphragm-type pressure-regulator valve.

In the present case, however, its application presented special difficulties, due to the necessity for a rapid, simultaneous and exactly corresponding adjustment of three different pressures; namely, one in the atmosphere inside the test-chamber, one in the engine intake-air feed, and one in the engine exhaust-gas piping.

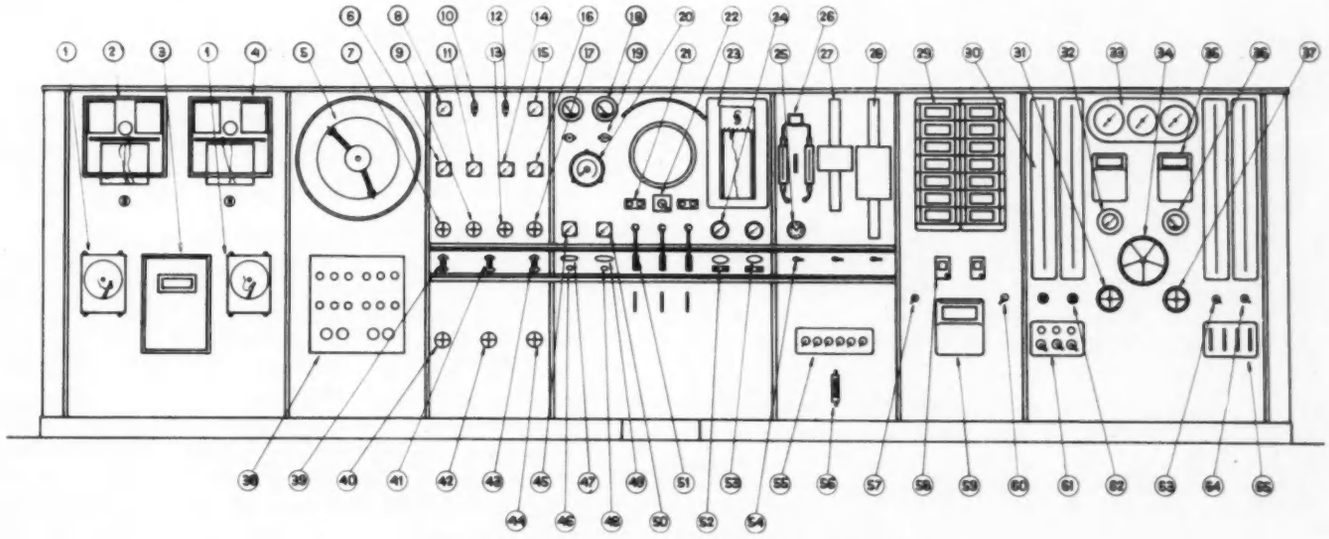


Fig. 19—Diagram of the Central Control-Board

(1) Altitude Recorder
(2) Temperature Recorders of the Cell Ambient Air and of the Engine-Feed Air
(3) Hygrometer
(4) Temperature Recorders of the Engine Oil and Water
(5) Engine Oil-Consumption Meter
(6) Engine Outlet-Oil Thermometer
(7) Engine Oil-Refrigerant Control
(8) Engine Inlet-Oil Thermometer
(9) Left Exhaust-Box Cooling-Water Control
(10) Exhaust-Box Cooling-Water Spy-Glass
(11) Left Exhaust-Box Cooling-Water Thermometer
(12) Engine Cooling-Water Spy-Glass
(13) Right Exhaust-Box Cooling-Water Adjustment
(14) Right Exhaust-Box Cooling-Water Thermometer
(15) Engine Inlet-Cooling-Water Thermometer
(16) Engine Outlet-Cooling-Water Thermometer
(17) Exhaust-Box Water Refrigerant-Control
(18) Turbo-Blower-Motor Ammeter
(19) Switch-Control Buttons
(20) Tachometer
(21) Oil-Pressure Spy-Glass for Engine and Brake Connecting-Shaft
(22) Engine-Ignition Switch
(23) Airflow Recorder
(24) Thermometer
(25) Chronometer
(26) Fuel Flowmeter; Direct-Reading Type
(27) Fuel Flowmeter
(28) Fuel Flowmeter; Volume Type
(29) Engine-Exhaust Pyrometer
(30) Barometers; Mercury Type
(31) Altitude Automatic-Regulator Control; for Slight Adjustments
(32) Oil-Pressure Gage
(33) Altitude Dials
(34) Altitude Automatic-Regulator Control; for Considerable Adjustments
(35) Thermometers; for Cell Ambient Air and Engine-Feed Air
(36) Thermo-Coil Ammeter; for Heating of Cell Ambient Air and Engine-Feed Air
(37) Thermo-Coil Rheostat
(38) Fuse Board
(39) Oil-Heating Thermo-Coil Switch
(40) Engine-Oil Cock
(41) Water Primer-Pump Switch
(42) Water Primer-Pump Cock
(43) Water-Heating Thermo-Coil Switch
(44) Engine-Water Cock
(45) Engine Oil-Pressure Gage
(46) Oil-Tank Depression-Control
(47) Oil-Tank Vacuum-Meter
(48) Fuel Float-Chamber Depression-Control
(49) Fuel Float-Chamber Vacuum-Meter
(50) Fuel Pressure-Gage
(51) Engine Controls
(52) Brake-Load Adjustment
(53) Brake-Load-Adjustment Dial
(54) Fuel-Flowmeter Cocks
(55) Fuel Cocks
(56) Fuel Spy-Glass
(57) Air Cock for Fuel-Vapor Warners
(58) Fuel-Vapor-Warner Release
(59) Warner of Fuel Vapors Inside the Hermetical Cell
(60) Air Cock for Fuel-Vapor Warners
(61) Altitude Adjustment-Cocks
(62) Engine Oil-Pump Cock
(63) Rapid-Descent Cock
(64) Compressed-Air Cock
(65) Thermo-Coil Switches; for Heating the Cell Ambient Air and the Engine-Feed Air

As a matter of fact, it would be absolutely impossible to operate satisfactorily, separately, by hand, the regulating valves controlling the three pressures concerned. This would cause, as it actually did in the first tests attempted with the test-plant, serious variations in engine output, and consequently in the speed of the engine, a condition which could not be permitted during engine tests of this nature. Simultaneous and rapid control of these three pressures just mentioned has been attained in the manner shown on the diagram, Fig. 10. In its definite form, the apparatus is situated behind the central control-board, and is operated by the single hand-wheel mounted on the control-board, as already mentioned.

Finally, the rapid-descent lever actuates a compressed-air controlling-device, which is connected to the automatic altitude-control previously described, and which allows intake air from the two cooling units to flow directly into the test-chamber. This consequently raises the pressure of the enclosed air; that is, it lowers the altitude. This is of course accomplished under strict control of the automatic altitude-regulator to which the rapid-descent lever is connected.

The Fiat High-Altitude Test-Plant is the result of designs which grew out of a long series of experiments and research work, and it is a matter of considerable satisfaction to the technical staff of Fiat to see this plant accomplishing day by day, in a most satisfactory manner, the practical work for which it was originally designed.

Discussion

Some Comparisons Made with American Practice

—Ralph N. DuBois
Lycoming Mfg. Co.

THOSE of us who have been concerned with the altitude testing of aviation engines are filled with admiration for the engineers of the Fiat Co. in the conception of such a project as that described by Mr. Merriman. The provision for altitude testing of air-cooled engines is especially interesting, since such equipment does not exist in this country. The effect of lowered density and temperature on air-cooled powerplants is certainly as worthy of investigation and study as on liquid-cooled equipment. The difficult problem of air cooling at reduced atmospheric pressure in a closed-circuit tunnel has apparently been solved in a satisfactory manner.

The capacity of the plant for testing air-cooled engines seems low for such a recent development. Air-cooled engines of 750 to 800 b.hp. are in production in this country and 1000 b.hp. has been developed in certain cases. It would seem that the air-cooled capacity of 500 b.hp. in the Fiat plant should be increased if the trend toward higher output for this type of equipment continues.

As regards the details of the plant, the provision of sufficient refrigeration capacity for cooling of the exhaust gases is certainly novel. In the altitude laboratory of the National Bureau of Standards, the exhaust is cooled by water-jacketed manifolds plus water spray, and the resulting mixture of exhaust gas and water vapor is handled by two Nash "Hytor" rotary vacuum pumps driven by 150-hp. electric-motors. The total power required is 300 hp., or the same as the Fiat exhaust system. The use of water and brine sprays for cooling of the intake air is also new to us. I am curious to know if means are provided to compensate for the pressure loss through the various air-temperature regulators, so that air may be delivered to the engine intake at standard atmospheric pressure. In this country, the Air Corps requires that calibrations be made with a positive pressure of 25 in. of water at the air intake to the engine.

The Fiat engineers are to be congratulated on their success with as many automatic devices as are described by Mr. Merriman. It has been the writer's experience with equipment of this nature that the addition of automatic control-devices follows the law of diminishing returns, reaching a point where the chief occupation of the test personnel is the servicing of the test equipment, rather than the conducting of tests.

It is to be hoped that Mr. Merriman's excellent description of the Fiat plant will serve to increase the interest in altitude testing of air-cooled engines in this country, to the end that similar equipment may eventually be provided by our Government.

Advances in the Development of Altitude Engines Presaged

—N. N. Tilley
Continental Aircraft Engine Co.

IT is interesting to note that a manufacturer rather than a governmental department is equipped to conduct altitude development of engines. Such complete equipment to duplicate any altitude condition to permit ready and more precise solutions of particular problems is the desire of all good engineering organizations. Satisfactory engines for a considerable range of altitude conditions can be guaranteed by manufacturers with such a laboratory. The action of the Fiat management in providing such facilities for the development of engines of superior altitude performance, operation and reliability, presages future advances in development of altitude engines as well as appreciable use of such engines.

To those without access to an altitude laboratory, the alternatives are flight tests with specially equipped airplanes or to determine separately effects of some altitude condition on each element or system of powerplant, such as power output, electrical system, fuel system, cooling system, and the like. Flight tests are used to check final conclusions in any event. Doubtless the Fiat engineers can disclose instances of elimination of errors that come from conclusions based on sea-level tests corrected to desired altitude condition by results of tests on elements, whether performance or operation features are under investigation.

A comparison of costs in man-hours of the various methods of obtaining altitude data would be of interest. The rapidity of securing necessary data for particular problems should be best for the complete laboratory and in many cases the costs should be minimum.

There is little that one can comment upon concerning the elements of this rather complete altitude laboratory other than that considerable foresight has been exercised to secure utility with safety.

The Fiat Co. is to be envied for attaining such a favorable position for development of altitude engines.

When Is a Truck Tire Overloaded?

By J. E. Hale
Firestone Tire & Rubber Co.

LENGTHY consideration is given to tire overloading, with the understanding that overloading as referred to in tire failures is quite different from the application of the term overload to structural materials which collapse under a reasonably well-defined excess of load.

While this paper deals primarily with overloading, there are so many other aspects relating to the use of truck-bus tires affecting the industry that a discussion is included of various other phases of the tire business intended to be instructive along the line of longer life and greater freedom from trouble.

The increasing varieties of service in which motor-vehicles are being placed demand different types and characteristics of tires, which are outlined. Then there is a discussion of the relative merits of the balloon type versus high-pressure-type tires.

The choice of tires for new trucks and buses is covered in a practical way and also there is a section outlining the variations of the basis for determining loads and air-pressure recommendations.

For the truck operator's benefit, there is included a section as to what the operators should know and practice to get the most out of tires, discussing the importance of inflation, dual mating, wheel alignment, repairs and retreading on both tires and inner tubes.

VERY early in the development of the pneumatic-tire industry the absolute necessity of maximum load ratings and recommendations for air pressures was realized, so that, for the last twenty-five or thirty years, we have had tire-load and inflation tables which are familiar to everyone. These were compiled by the Tire and Rim Association, long recognized as the standardizing body representing the tire manufacturers. The tables are set up to show the maximum limit of load beyond which the tire is not supposed to be loaded, and for this load an air-pressure recommendation is also included. In spite of repeated isolated insistences, either verbal or in writing, for permission to exceed the maximum load or use lower pressures, liberties are rarely taken with the figures specified. Commercial guarantees on tires are based on these industry standards and tire company adjusters use these limits as a basis of discussion with customers in the event of claims for improper service of the tires.

The purpose of this paper is to review the technical, commercial and practical aspects of the subject of loads to which a tire should be subjected, in an endeavor to present logically the pros and cons of underloading and overloading for the benefit of those who feel that the load ratings are now too conservative. In the last analysis, it is the truck owner whom we have to satisfy, so it is proper that we should get before us clearly at this time the truck-owner's viewpoint.

I believe he wants, first of all, economical tire mileage, which is obviously measured in terms of cost per tire mile or cost per ton mile of goods carried. He is also interested in first cost. He is interested in a tire which has the proper traction and non-skid characteristics; a tire which can use low pressures to cushion his truck and his load; a tire which will wear uniformly and be free from premature failures of such a nature that they interfere with his continuity of service; a tire which is efficient and does not require the use of excessive fuel; a tire which will stand a reasonable amount of abuse and not fail; a tire which is resistant to punctures. Some of the foregoing features mentioned are functions of the number of plies; some are functions of the thickness and character of the tread stock; some are dependent on the character of the non-skid design; and some are functions of the size of the tire.

An outline drawing, Fig. 1, of the 9.00-20 truck-bus balloon-tire, with the important features of the tire construction labeled, is included for reference in identifying and differentiating the conventional terms, nomenclature and dimensioning practices commonly used among the tire designers.

The first part of this paper will treat with tire overloading in detail, as the tire designer sees the causes and effects. To begin with, we should first agree on our descriptive terms by simply stating that what we call "natural wear" occurs when the tire wears down through the tread without any interference in service from any other cause; while "premature failure" causes a tire to go out of service with the tread not completely worn out, on account of ply failure, separation, bead failure, and the like.

All truck operators have had impressed on them the customary warnings about overloading. Probably the majority of operators are acquainted with the results of overloading in

[This paper was presented at the General Transportation and Maintenance Meeting of the Society, Chicago, Oct. 10, 1935.]

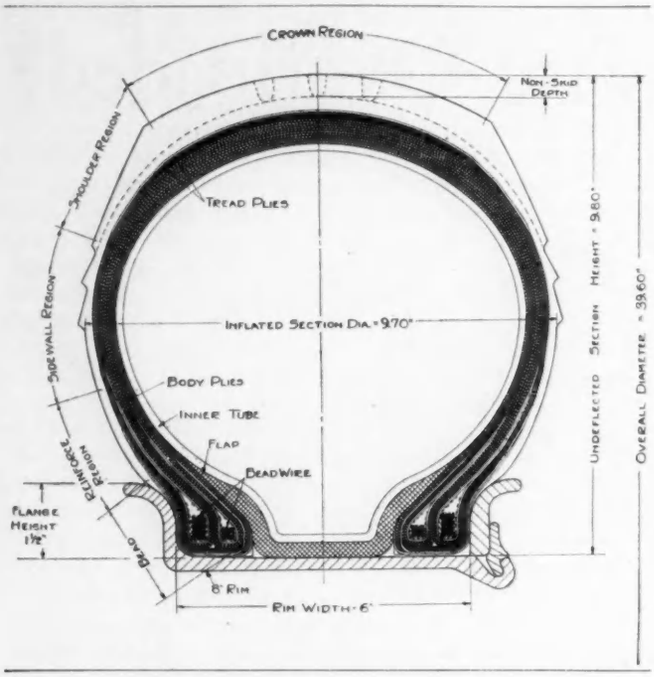


Fig. 1—A 9.00-20 Truck-Bus Balloon-Tire, Tube, Flap, and Rim Assembly, Showing the Principal Elements and the Points of Measurements

a general way, but at the same time they really know little or nothing concerning its ultimate nature; so I will list a condensed outline of the tire-designer's classification of the conditions observed in tires which have gone out of service prematurely as a result of overloading. There are three general classifications of cause and effect:

(1) Excessive flexing of the cord-body structure fatigues the cords so that, finally, they weaken to the point where they develop into what is known as a "flex break".

(2) The greater the load on any tire, the greater is the rise in temperature level in the body of the tire due to dissipation of energy through molecular friction of the rubber and cotton. The result of the increased temperature is to bring about definite deterioration in both the cotton and rubber stocks and also to change the physical characteristics of both the rubber and cotton so that tires in the heated condition are particularly susceptible to "separations" and, under rough-road conditions, are likely to develop what we call "impact breaks".

(3) Increased loads mean increased physical stresses in various parts of the tires, particularly in developing a shear tendency which, in addition to the heat mentioned in item (2), contributes to the problem of "tread separation". These extra stresses develop more abrasive action against the road surface, resulting in faster tread wear; and finally, increased stresses developing in the bead region have a tendency to produce weaknesses at the point where the tire is attached to the rim.

Here are a few simple fundamentals which will state the technical aspects of how pneumatic tires function as load carriers. It is the compressed air inside the casing that carries the load. Strictly speaking, the casing carries part of the load due to its inherent stiffness. As a general figure to keep in mind, it is probably true that 75 or 80 per cent of the load is carried by the air and 20 or 25 per cent by the casing. Of course, the ideal tire would be the one in which the air carried 100 per cent of the load. Obviously, the greater the number of plies, with thicker sidewalls, the stiffer is the casing, and the less opportunity the air has of functioning; particularly is this true as the tire section gets smaller. For this reason, the balloon-type tires with larger section and fewer plies are the more desirable tire equipment.

With air under pressure inside the casing, it automatically deflects until the tread of the casing spreads out over a sufficient area to account for enough square inches of contact so that, theoretically, the air pressure multiplied by the area will equal the load. If our tire were a perfect torus and perfectly elastic, for instance like an inner tube, experimental loading would demonstrate the exactness of this relationship between the area of contact, the air pressure and the load. It happens that on account of structural effects, tread-design profiles, support of the rim flanges and the like in the tire, there is a slight deviation from the theoretical, but the statement is sufficiently true to justify using it as a generality.

For example, the 9.00-20 truck-bus balloon-tire, loaded to its rated maximum of 3250 lb. with the recommended pressure of 65 lb., settles down or deflects until it establishes a total area of contact of 45.4 sq. in. Fig. 2 shows an actual tread imprint. Now, if we doubled the load with the same air pressure, we should theoretically double the area of contact. As a matter of fact, the result from actual experiments is of the order shown in Fig. 3, showing the imprint of the ame tire with a load of 6500 lb. imposed on it.

The next phase to emphasize is the factor of tire deflection. It is customary to express tire deflection in percentage as a ratio between the actual deflection of the tire and the actual cross-section of the tire taken at a point not under load. Fig. 4 shows a cross-section outline of the tire loaded to 3250 lb. with an air pressure of 65 lb. The dotted line enables us to demonstrate the actual deflection which, expressed as a percentage of the actual section, gives a deflection of 11.4 per

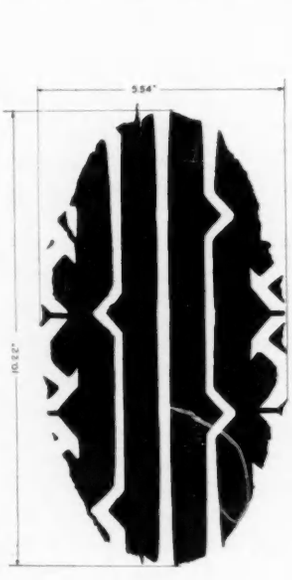


Fig. 2—An Actual Tread Imprint of a 9.00-20 Truck-Bus Balloon-Tire

Load, 3250 lb.; air pressure, 65 lb. per sq. in.; total area, 45.4 sq. in.; actual area, 33.8 sq. in.

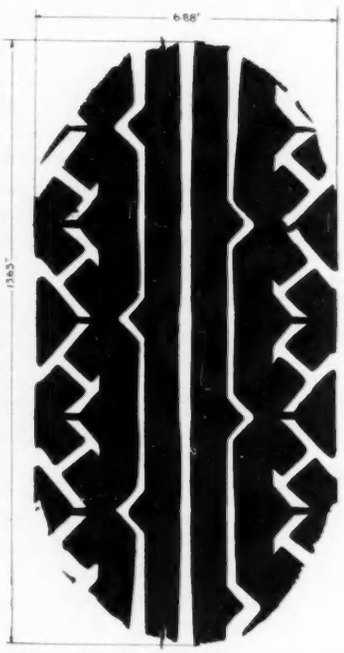


Fig. 3—The Imprint of the Same Tire Shown in Fig. 2 with a Greatly Increased Load Imposed on It

Size, 9.00-20; load, 6500 lb.; air pressure, 65 lb. per sq. in.; total area, 81.6 sq. in.; actual area, 63.7 sq. in.

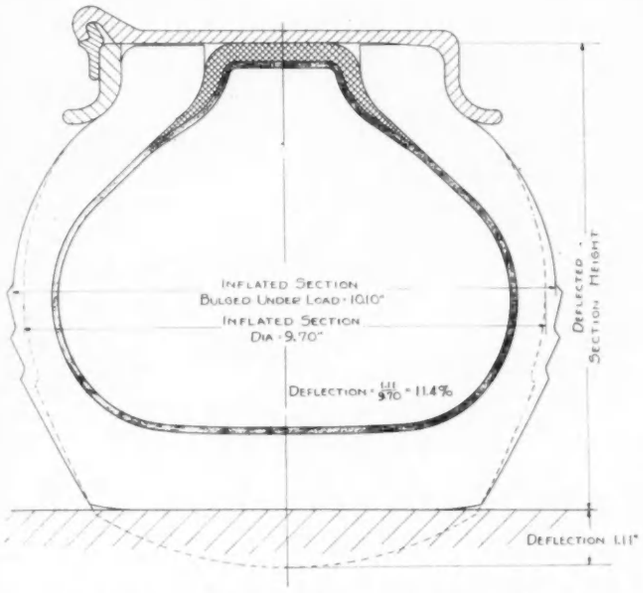


Fig. 4—Cross-Section of the Tire Shown in Fig. 1 When Loaded

The load was 3250 lb. and the air pressure was 65 lb. per sq. in.

cent. In a second case, Fig. 5, where the load is doubled to 6500 lb., the relationship between the full lines and the dotted lines shows the percentage of deflection to be 21.5. This feature of deflection is all-important, in that it determines the smallness of the curvature of the tire body at the point of maximum bulge. This would not be important under static conditions, but if a condition existed which caused alternate loading and unloading of the tire as it stood in one position, flexing the sidewalls from normal contour, Fig. 1, to a deflected condition as shown in either Fig. 4 or Fig. 5 and back again, obviously there would occur a gradual fatigue of the materials of the sidewalls after a great number of repetitions of these flexes. It is also obvious that the degree of flexing is greater where the deflection is greater, as shown in comparing Figs. 4 and 5, and the result would be that the condition shown in Fig. 5 would develop more rapid deterioration than that in Fig. 4.

A truck in motion loads and unloads the complete circumference of each tire once with each revolution of the wheel. Since in traveling 35,000 miles the wheels of an average truck make 15 to 20 million revolutions, it will be seen that the sidewalls of each tire must flex that many times without failure. In Figs. 6 and 7 are shown "flex breaks" in tires which have failed by excessive flexing. As mentioned previously, this is the result of a fatigue of the cord. Flexing of the tire not only flexes the cords, but alternately stretches and upsets them twice for each wheel revolution. First the ply yarns of the cords break at isolated spots along the circumferential zone of greatest flexing. Miles later the cords break through, and still later the individual breaks increase in numbers until finally the casing is weakened sufficiently to blow out.

The foregoing discussion has been presented on the basis of loads, air pressures and deflections under control; as, for instance, in rolling over perfectly smooth, level roadways, assuming the deflections would be as when observed and measured under static conditions. In actual truck operations, the road surface is uneven; the result is a jouncing up and down with instantaneous deflections much exceeding those observed statically. In addition to up-and-down action from jouncing, we have to keep in mind the lateral stresses imposed on the tires in going around curves, lurching sidewise against obstacles in the road, and so on. This latter action, in combination with the jouncing, is of course the most destructive action of all.

The number of plies used in a tire is determined by three factors: the air pressure to be used, the section diameter of the tire, and the fact that experience has taught us that on account of the bursting stress of compressed air, if a tension of 2 to 4 lb. per cord is exceeded, the life of the tire will be shortened. From this set-up we are able to compute the desirable number of plies for all combinations of tire sections and pressures, and the results are shown in Fig. 8.

We are now ready to emphasize why and to what degree there must be a limit of vertical deflection. By referring to Figs. 4 and 5, it will be seen that the curvature of the sidewall at the point of bulge has a much larger radius in the case of the 3250-lb. load than in the case of the 6500-lb. load. It is very obvious that, in the event of even greater loads, the radius of curvature at the point of bulge would be reduced to the point where an effect in the nature of a crease in the sidewall might develop; for example, the jouncing up and down of a severely overloaded truck operating over a very uneven road would momentarily, at least, produce a very pronounced effect of this character. Our studies have made it clear that the radius of curvature can be lessened by using fewer plies. It also works out that the larger the section, the less tendency there is to approach a limit on smallness of curvature at the point of bulge. The whole situation sums up to a range of deflection limits dependent on number of plies and cross-sections as shown in Fig. 9.

The final phase of design and physical characteristics to be discussed is the tire cross-section and overall diameter. Our basis, of course, is a given load per wheel and an air pressure which is considered desirable to provide the necessary riding comfort, protection to the vehicle, and the like. This com-

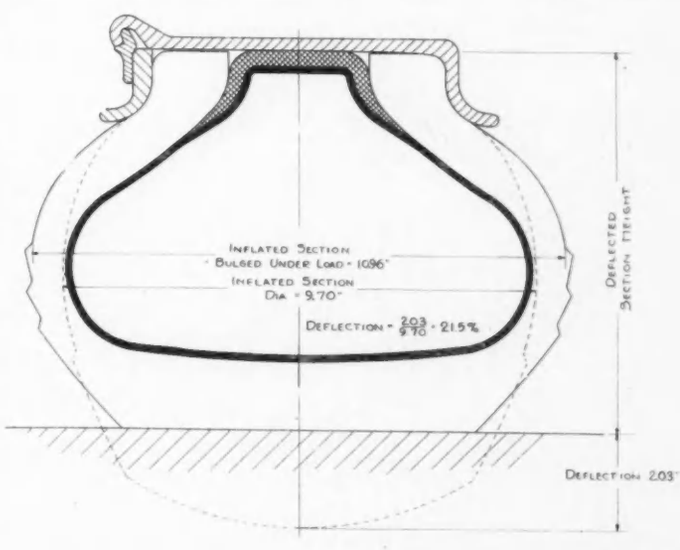


Fig. 5—Cross-Section of the Tire Shown in Fig. 1 When the Load Is Doubled

The load was 6500 lb. and the air pressure was 65 lb. per sq. in.; or, the same load as for Fig. 4 (3250 lb.) with an air pressure at 32.5 lb. per sq. in.

Figs. 6 and 7—"Flex Breaks" in Tires Which Have Failed by Excessive Flexing

bination determines the area of contact. The area of contact, in turn, determines the section of the tire, the overall diameter and the limits of deflection. The limits of deflection are determined by the tire section and the number of plies. The necessary variables in this group can be set down in an equation from which we can derive the relationship between load, tire section and overall diameter.

Formula for Computing Load Rating of Truck-Bus Balloon-Tires.—I give, in the following, an empirical formula which has been developed from our years of experience in tire use, and which formula is the basis of our load and inflation tables:

$$W = 0.425 \times P^{0.585} \times S^{1.39} \times (D_r + S)$$

where

W = Tire-load rating in pounds
P = Inflation pressure in pounds per square inch
S = Actual section diameter in inches
D_r = Rim diameter in inches

While, in the foregoing, stress has been put on the limit of deflection of the casing as a controlling factor in limiting tire life, there are other important factors of tire performance

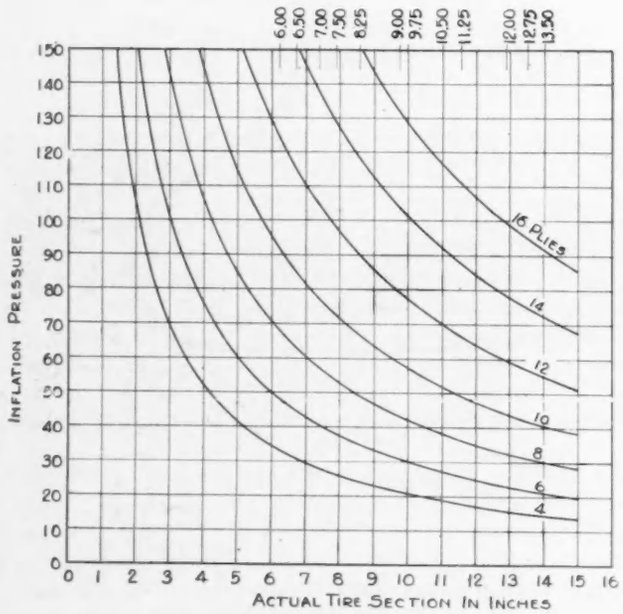


Fig. 8—Chart Showing the Maximum Inflation Pressure for Tire Cross-Sections of Various Numbers of Plies

on which I will touch briefly. First, there is a type of break in the body plies under the tread which we classify as an "impact break". The tire may have seen relatively short service; yet if the load, the speed and the heating effect is just right when the tire strikes an obstacle or hole having a protruding edge, the result is an instantaneous rupture of cords through all plies. This type of break may go diagonally across the crown of the tire or may take the form of a large X; see Figs. 10 and 11.

I would like now to mention the effect of overloading in producing tread wear and developing tread separation. By referring to the tread imprints in Figs. 2 and 3, it will be seen that, as the load is increased, thus producing a much larger area of contact, obviously there is more scuffing action as the tread lays down on the pavement and picks up in rolling around. The result of this is faster tread wear. It is difficult to show in a drawing, although, by referring to Figs. 4 and 5, this scuffing action may be sensed; also, the tendency to push the shoulder of the tread in a shear action off the body plies is much more pronounced. This action is particularly aggravated on rough pavements where high bricks or cobble stones protrude in a manner which would develop high intensity of pressure at localized spots in the shoulder of the tire. The result is that, after many repetitions of this action, the cushion stock may actually be sheared from the body plies in a small area. Then it is just a question of time until this separation spreads and ruins the tire.

The deteriorating effect of heat on rubber and cotton has already been mentioned. Our investigations have shown that the points of maximum flexing and movement of rubber stocks are the points of generation of maximum heat, and it is in the upper sidewall and shoulder region of the tire that this effect is most pronounced. In the larger tires, having 12 and 14 plies, it is known that temperatures of 200 to 275 deg. fahr. build up during hot summer days. Any rubber stock in a heated condition has less tensile strength and is what we call "short", and in that condition much more likely to yield under lateral stresses as in shear.

In order to show more effectively the heat build-up, the temperature rise as influenced by four variables is plotted in Fig. 12, assuming a given tire and using the variables of inflation, atmospheric temperature, load and speed. Keeping the speed, load and inflation constant, as the atmospheric temperature increases, the tire temperature goes up rapidly. Keeping the atmospheric temperature, load and inflation constant, as the speed increases, the tire temperature goes up

rapidly. Keeping the atmospheric temperature, speed and inflation constant, the tire temperature goes up rapidly as the load is increased. Keeping the atmospheric temperature, speed and load constant, the tire temperature drops rapidly as the inflation is increased.

Since the effect of variables in developing heat has been mentioned, we ought also to include a brief reference to heat and pressure build-up starting from a cold tire, as when the truck has not been operating for several hours or over-night. In Fig. 13 the heat and pressure build-up figures from a cold tire are very clearly indicated.

An overloaded tire on an undersized rim is often troublesome. The tire beads are crowded together in an unfavorable position, and in most combinations the rim flange is not high enough to support properly the lower sidewall. The result is too much "bead-rocking" action. Chafing develops either externally at the region of contact with the rim or internally around the bundle of bead wire. Presently, the cord plies all pull loose from the wire and the tire is done for. Often this brings about rim trouble, since the rim, not being designed for use under such extreme conditions, becomes fatigued from repeated stress action. This is particularly true in cases where the operator resorts to over-inflation of the tire to make up for the overload.

Another important question which is often brought up is the effect of overloading tires occasionally for short periods of time, as in the case of a truck in which the loads are such that 90 per cent of the operation is within the rated capacity of the tires, but 10 per cent of the time the truck is overloaded. The only way we can answer this is to point out that overload takes something out of the tires. If the overload is only for a very small percentage of the operating mileage, and the overload is only very moderate, what it takes out of the tires possibly would be hardly noticeable. On the other hand, the larger the percentage of operating mileage which is overloaded, and particularly the greater the percentage of overload, obviously, the more the tires will suffer.

We tire people realize that an operator has a truck that he has to make deliveries with and that on some of these deliveries it is expedient for him to carry whatever loads he has, even if he is conscious of overloading. We certainly do not countenance such a practice. All we can do is to urge, if it has to be done, that the load be distributed as much as possible over all the tires on the truck so as to even up the overload; that the truck be operated at slower speeds to keep down the heat and danger of impact breaks, particularly on uneven pavements; and, finally, that the tires be properly inflated to give them the best possible chance to avoid excessive flexing.

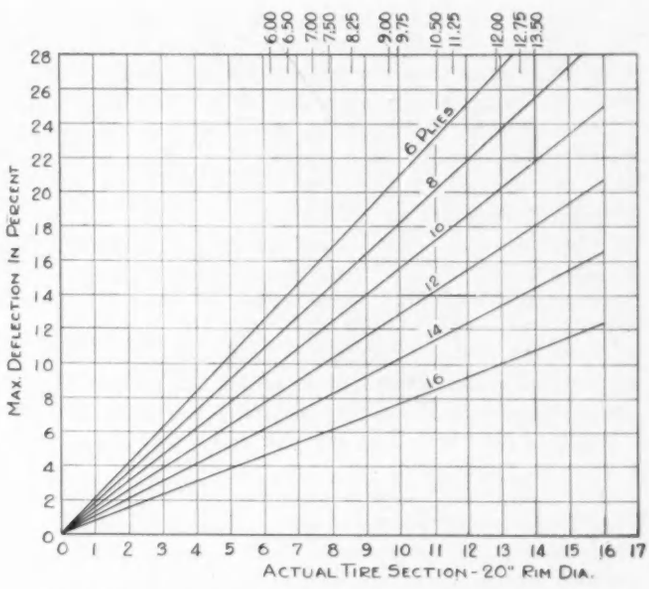
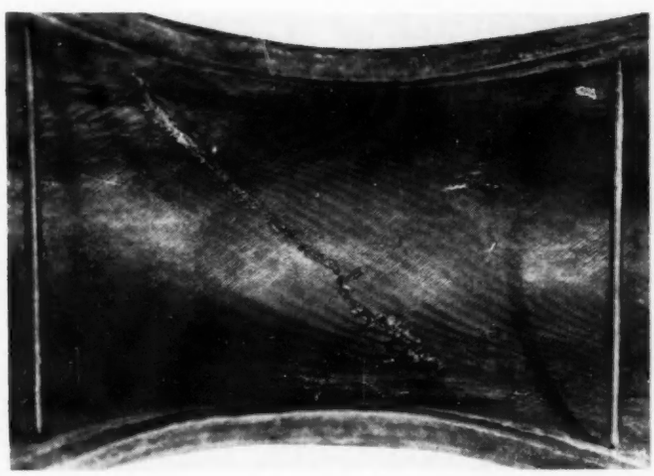


Fig. 9—Chart Showing the Limit of Deflection for Tire Sections and of Various Numbers of Plies

At this point it is probably wise to inject some comment about over-inflation. We occasionally find truck operators who, by some experimenting on their own initiative with over-inflation, believe they have made a discovery which justifies their going ahead with overloading. We realize that there may be an occasional case of an operator whose conditions just happen to be the right combination of favorable circumstances to allow him to work this out on the positive side; but by far the majority of cases would be so definitely on the negative side that we cannot for a moment agree to resorting to over-inflation as a means of carrying more load. On the positive side is the tendency to get out of flex-break trouble. This is the only helpful aspect. Against this, we find that



Figs. 10 and 11—Illustrations of Instantaneous Ruptures of Cords Through All Plies
This type of break may go diagonally across the crown of the tire as in Fig. 10 or may take the form of a large X as in Fig. 11

the stress per cord is increased sufficiently so that impact breaks immediately become as annoying as flex breaks were with the overload and normal inflation combination. In addition to impact breaks, there are likely to be bead failures and rim failures. Also, under conditions of uneven pavements and high speed, tread separation is excessive, and there is a very decided tendency to develop tread cracks in the grooves between the non-skid elements.

While the term overload is commonly used in engineering, its significance may or may not be properly understood when reference is made to specific structures or machines. Nearly everyone understands that an overloaded beam breaks or collapses, and that an overloaded boat sinks. The effect of an overload on an electric motor or gasoline motor is understood at least by technical people; but I feel that the understanding of the effect of an overload on pneumatic tires needs a clear-cut presentation.

In this discussion, keep in mind that we have tremendous variations in combinations of speed, heat, load, road-surface irregularities, and air pressure, which make it impossible to establish for tires anything like a factor of safety as in the case of stresses in beams, columns, and the like. Obviously, the different types of tire wear and failure further complicate the situation; so that about all we can do is to report the effect of load as a trend. The best way of doing this is by means of a plot displaying the average-life mileage of tires plotted against load; see Fig. 14. For example, take the 9.00-20 at its rated load of 3250 lb.; the plot shows a life of 35,000 miles. Now, suppose a duplicate tire to have been overloaded 100 per cent (6500 lb.); the plot shows the tire out of service at 7500 miles.

In a way, Fig. 14 is the most important feature of this paper, but not as a reference work for settling indisputably the life mileage of a tire on which one could depend. Road-surface conditions, variations in speeds, seasons and climates, neglect or abuse, air pressure and difference in brands of tires, add up to such an infinite number of variables that all we can do is to offer this plot as a fair and reasonable trend based on our experience.

The curves in Fig. 14 indicate that tire mileage varies approximately inversely as the square of the load. It will also be noticed that the plot is subdivided into three zones which are labeled; trouble-free zone, zone of occasional blow-outs, and zone of frequent blow-outs. There is no sharp line of demarcation between these zones. Obviously, they would graduate from one into the other; but the intent is to convey

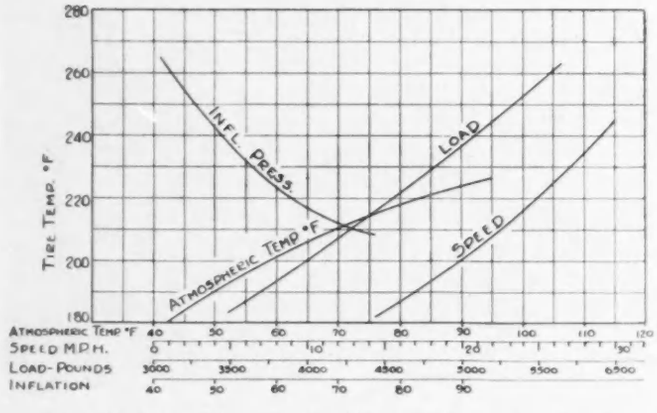


Fig. 12—Chart Showing the Effect of Increasing Load, Speed, Inflation and Atmospheric Temperature on Tire Temperature

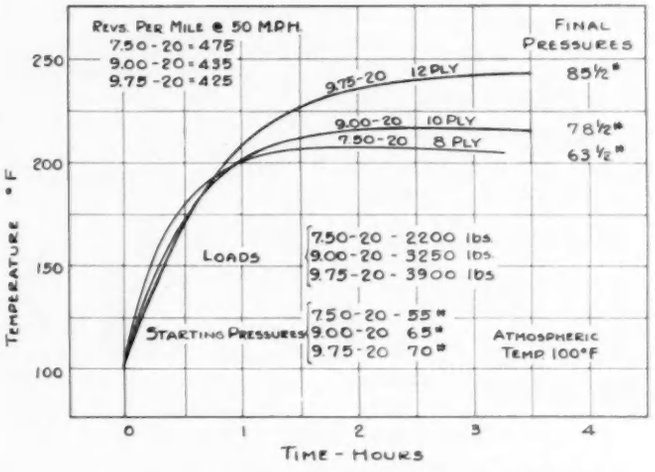


Fig. 13—Chart Showing the Temperature and Pressure Increase, Operating at a Constant Speed of 50 m.p.h.

the idea that, as tires are operated under heavier and heavier loads, the durability of the tires becomes more and more seriously impaired from the interference of body-ply failures which take the tires out of service prematurely.

The first thing to notice is that the plot shows no breaking point as in the stress-strain diagram of steel, but rather a gradual diminishing of life as load is increased; consequently, there is no point corresponding to elastic limit or yield point. Just where on these curves would we decide to label the point of overloading? I am sure any decision would start plenty of argument. One thing is obvious; it is that a truck operator could write his own ticket in tire mileage if he would reduce his load per tire as indicated in the plot. As a matter of fact, public usage has developed what I call "mileage expectancy." All operators have in mind their own average performance which they expect to secure, and if the tires do not deliver it, the owners are very critical of the particular make of tire in question. Year after year this mileage-expectancy figure has gone up gradually, until now owners expect anywhere from 25,000 to 40,000 miles average.

I am including a plot in Fig. 15 as a means of showing the trend of progress in improving the durability of truck-bus tires; and I think, considering the fact that tires of this type were practically unknown until long after passenger-car tires were pretty well developed and that it was only around 1915 when they began to get into any reasonable production, the tire companies have made really commendable progress in perfecting a product which is subject to such extreme conditions of use.

This increase in durability has been the result of four factors: first, the gradual improvement in the tires themselves as the tire companies have made progress in perfecting their products; second, the increasing effectiveness with which the truck manufacturers study the truck-buying prospect's operating conditions and urge choice of adequate tire and rim equipment on new trucks; third, attention on the part of owners to proper inflation and care against injuries and abuse; and fourth, much improved highways, and also a better understanding of the economies of adequate tire size. The tire companies have, of course, been largely instrumental in bringing this progress about because costly experience has taught them the absolute necessity of educational work with operators and manufacturers, designed to get the most out of tires.

Truck and bus engineers have been grumbling a little because the tire industry has not seen fit to revise its load schedules upward. They think we are too conservative and are arguing that many a truck is overloaded now, and that in consequence we should bring our ratings more in line with what is actually being used by the operators. Such a contention, on the face of it, appears to have merit. If it were to be put into effect, it would be a simple question of the industry's going out, weighing loads and just putting in the necessary amount of effort to determine the actual average figures on tire loading as it is practiced and then constructing our tables.

I am sure that the truck and bus industry will find the tire manufacturers willing to undertake a thorough discussion of this subject; but it is very clear to me that we could not afford to consider any such step as that of a revision of tables to bring the load ratings up to the point of the average loads as they are now carried, because of the psychology surrounding the whole situation. We tire people believe the truck industry and truck operators are going to discount any table which is printed. We must have in effect a table which is conservative enough so that, if followed religiously, it will in a very large majority of cases insure the tires a chance of wearing out by natural wear and deliver a life mileage substantially in accordance with the mileage expectancy on the part of the public. We feel that if the load ratings were increased materially, truck manufacturers would immediately start to sell their trucks equipped with tires in sizes too small, thereby saving themselves some money on the original cost of the truck, even enabling them to lower their list prices to a certain extent. Certainly, any such move would be widely talked about and the truck owners would also get into the spirit of the thing and they would soon be in the frame of mind where they would feel that with a size or two smaller they could carry the same loads they had been carrying and still expect the same mileage. A move of this kind would unquestionably lead to an increase in premature tire-failures, and as such failures develop into requests for adjustments, the tire companies would have no alternative but an increased activity in advocating oversizing on the trucks which were in trouble.

I can only emphasize that tires as now made by reputable companies are in a certain state of perfection as to what they will do under given loads and average operating conditions, and refer again to the plot in Fig. 14 as presenting the relationship between tire mileage and load. Fig. 15 showed

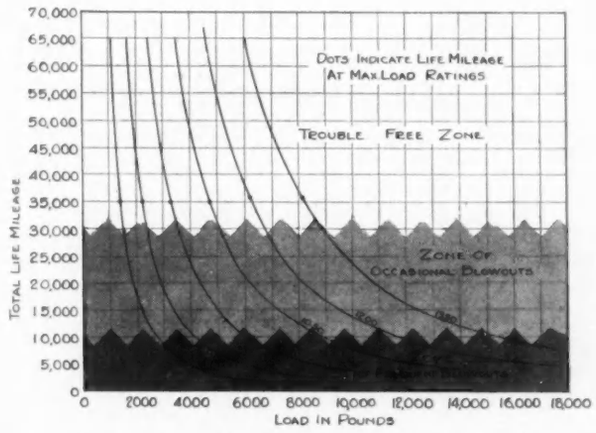


Fig. 14—Chart Illustrating How the Total Life Mileage of a Tire Is Reduced as Load Is Increased

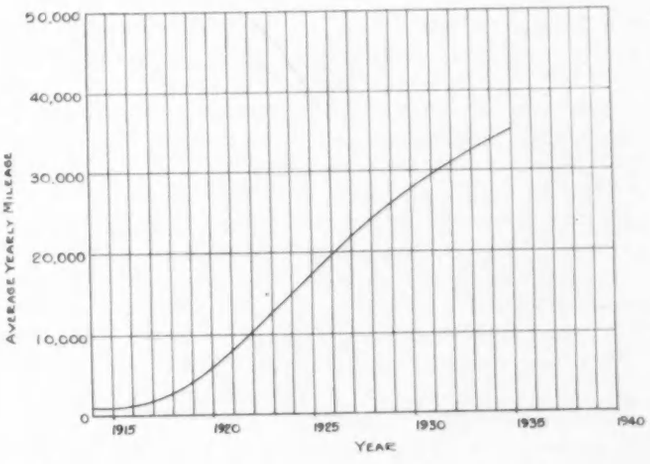


Fig. 15—Illustration of the Year by Year Trend of Increase in Life Mileage of Truck-Bus Tires

the progress in gradually improving tire durability. We tire people have every reason to feel that as time goes on we will be able to extend the curve upward in a manner which will mean greater durability. However, if our load-rating tables are arbitrarily increased too much, I anticipate that there will be a sharp drop in this durability line. We have no magic by which we can change this plot over-night by changing some load-rating tables; and if changing the tables brings about the use of smaller-size tires to carry the loads now handled by operators, the tire mileage will certainly go down.

The preceding discussion pertains to tire-life durability and it is interesting to interject at this time a little presentation of the cost-per-mile aspect, so I am including a plot, Fig. 16, which shows that, as the tire sections are increased to carry certain loads, the cost per tire mile decreases. This is very important, economically, particularly for the most progressive truck owners who study costs on a unit basis.

The foregoing discussion may sound like propaganda indicating reactionary resistance on the part of tire manufacturers to progress. It really is not. Our experiences in actual commercial applications have made us wary of yielding too easily to the demands for a move dictated solely by technical considerations. The commercial aspect must be emphasized. Since all tires sold are subject to adjustment, the adjusting end of our business simply must have some sound, fair, clear-cut basis on which to adjust claims. It has been the intention of the tire companies to set the load ratings at such a point that, if the tires are used under the conditions indicated, they will either deliver complete satisfaction or can be considered as having failed because of imperfect workmanship; but not because of lack of thoroughness in design or the proper selection of materials. Consequently, tire manufacturers are of necessity bound to resist pressure toward any changes which may lead us into obvious difficulties.

On the other hand, we are open-minded on the matter of upward revisions of load ratings. It is recognized that there are some inconsistencies in our tables; that the export situation, as mentioned later in this paper, must be taken into consideration; and also that there are some benefits which would accrue to the motorbus manufacturers if the tables could be revised to a certain extent. The Tire and Rim Association made slight revisions upward in its tables this year; as a matter of fact, I was the one who was instrumental in getting this subject opened-up. Just how much farther we can go

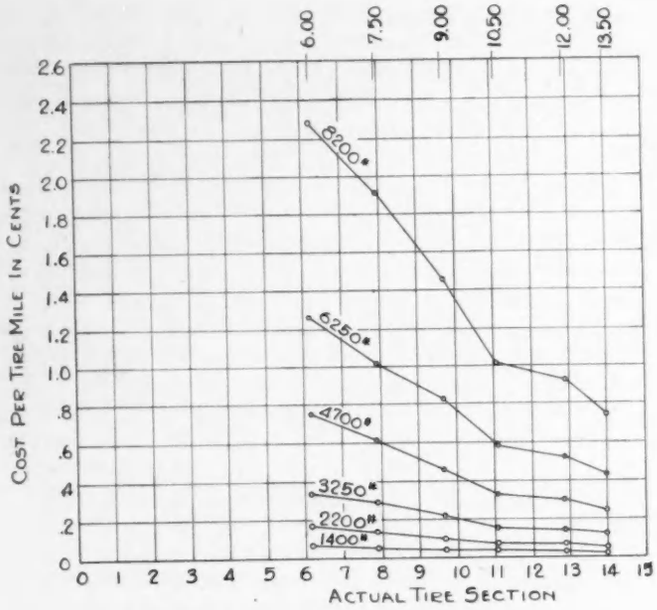


Fig. 16—Showing the Economy of Large Tires in Bringing Down Unit Costs
The chart is based on the data in Fig. 14

along this line can only be decided after very thorough consideration.

The familiar Tire and Rim Association tables are not reproduced here, but those for the truck-bus balloon lines are shown in plot form in Fig. 17. The abscissas are taken off at the actual industry maximum cross-section, since obviously there is no sense in plotting tire names against loads. It will be noted that the curves are not smooth, thus indicating an irregular step-up in load capacity as the sizes increase. As a matter of fact, the sections of the tires do not step up uniformly from one size to another; that is, the increment in cross-section size, while very regular from one tire size to another in the name size, is irregular in actual measurements. To show this, I am including Table 1, in which Column *F* shows the nicely uniform step-up in name sizes, while Column *G* shows the very irregular step-up in actual section measurements. As an example, note that there is 1.10-in. step-up from the 8.25 to 9.00; only 0.40-in. from the 9.00 to the 9.75; and so on.

A tire is useful only when it is mounted on a rim; consequently, the width of the rim is a factor in the tire measurements. The rim and tire combinations used in the truck-bus balloon line are the result of arbitrary choice of existing rims as the balloon sizes were added to the line, one at a time. In order to show the numerical relationship, Column *D* in Table 1 is added, in which it will be seen that the ratio of rim width to tire section varies from 55 to 66 per cent. While opinions may differ, our company feels that the ideal situation would be to have the rim width 62½ per cent of the tire section, which is a good average of existing practice. If we take this 62½-per cent relationship, with the tires as they are now designed and in use, and work out a new line of rims, the calculated cross-section of the tires on the new rims would be as shown in Column *E*, and the step-ups from size to size would appear as shown in Column *H*. From this it will be noted that in one case the step-up from size to size is only ¼ in., whereas, in another case, the step-up is 1¼ in.

Now taking the existing tires on a calculated line of rims of 62½-per cent section, we can plot these tire sections, Fig. 18, against the loads now carried and get still another picture of the inconsistencies of our load ratings on existing tires. Obviously, the 9.75, 10.50 and 11.25 tires are rated to carry higher loads than they should if the smaller sizes and larger sizes are correct. On the other hand, if the 9.75,

Table 1—Truck-Bus Balloon-Tire Line-Up; Size Relationships

A	B	C	D	E		F	G	H
	Tire and Rim Association			Calculated Rim Width and Cross-Section Using Same Tires Col. C on Rim 62½% of Section		Step-Up Size to Size		
Name Size	Rec. Rim Width	Max. Cross-Section	Ratio: Rim Width to Cross-Section (B over C)	Rim	Cross-Section	Name Size from Col. A	Actual Cross-Section from Col. C	Assuming Uniform Rim Width of 62½% from Col. E
6.00–20	3.75	6.20	0.605	3.91	6.26			
						0.50	0.55	0.78
6.50–20	3.75	6.75	0.555	4.40	7.04			
						0.50	0.70	0.62
7.00–20	4.33	7.45	0.582	4.79	7.66			
						0.50	0.50	0.26
7.50–20	5.00	7.95	0.629	4.95	7.92			
						0.75	0.65	0.92
8.25–20	5.00	8.60	0.581	5.52	8.84			
						0.75	1.10	0.89
9.00–20	6.00	9.70	0.619	6.08	9.73			
						0.75	0.40	0.57
9.75–20	6.00	10.10	0.594	6.44	10.30			
						0.75	1.00	0.52
10.50–20	7.33	11.10	0.660	6.77	10.82			
						0.75	0.50	0.70
11.25–20	7.33	11.60	0.632	7.21	11.52			
						0.75	1.35	1.24
12.00–20	8.37	12.95	0.646	7.97	12.76			
						0.75	0.60	0.84
12.75–20	8.37	13.55	0.618	8.50	13.60			
						0.75	0.55	0.75
13.50–20	8.37	14.10	0.594	8.96	14.35			

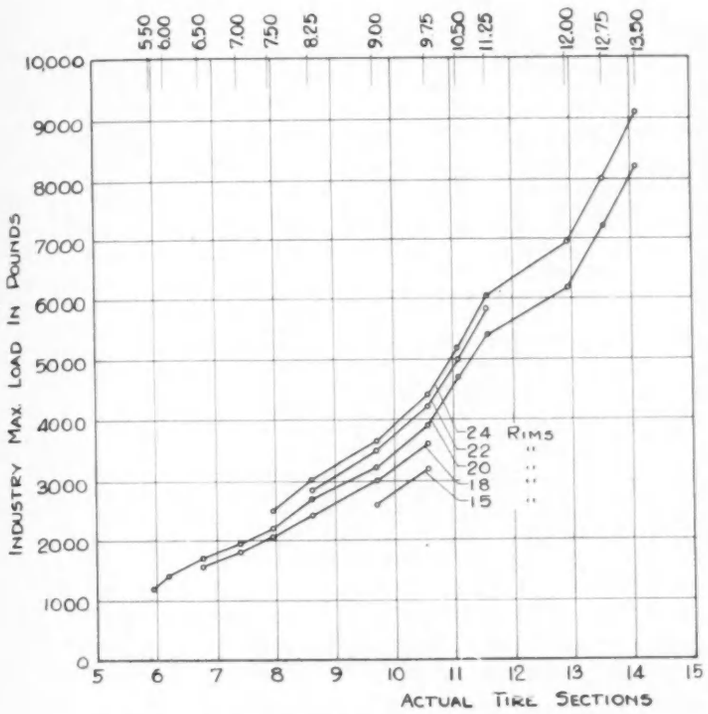


Fig. 17—Chart Showing the Tire and Rim Association Maximum Load Ratings for Truck-Bus Balloon-Tires

10.50 and 11.25 load ratings are correct, it is obvious that we could increase the rating on the sizes from 12.00 and up and 9.00 and lower.

While domestic truck operators have no knowledge of, or are not concerned with practices in export markets, it is proper for us to include in this review the status of load-inflation tables in foreign markets. For many years it has been the practice of American tire manufacturers selling in foreign markets to use and print the same standards of loads and inflations as standard for domestic markets. The one exception has been England, where their load tables have been slightly higher than here. The Michelin Tire Co., with headquarters in France, has for several years made a line of truck-bus balloon-tires to substantially the same size and dimensional charactertisics as our domestic line-up; but they have sold their product accompanied by a load-rating schedule averaging about 10 per cent heavier in the smaller sizes and with air-pressure recommendations about the same as ours.

The average of operating conditions in export markets where these tables are used is somewhat more favorable than in our domestic markets. Trucks in general operate at slower speeds, the overloading is not so great as in the United States, and the mileage expectancy is not so high. The plot in Fig. 19 gives a visual comparison of our schedules with the Michelin ratings. Until recently, American tire companies have not felt handicapped in selling in export markets under this status, but gradually Michelin's schedules have been given more prominence, particularly in some markets. A few months ago the large American companies found it expedient to print ratings for load and air, meeting those of Michelin. The result is that we make identical tires that have one rating when used in domestic markets and another rating when offered in export markets. We do not like this situation, but had to make the move as it was dictated purely by marketing conditions, there being no possibility of writing a world-wide standard of loads and inflation on account of the refusal of the Michelin company to participate in any standardization activities of any kind. Dunlop of England and Continental of Germany, on the other hand, have always shown great willingness to cooperate.

In presenting this paper, I have elected to refer only to the balloon type of tire and made no mention of the high-pressure tire. I did this purposely because I personally feel that there is no place for the high-pressure line in our present-day production of modern vehicles. While I recognize that thousands of trucks in use are equipped with high-pressure tires which will have to be replaced, it seems to me that the truck, bus and tire industries should be definitely active to see that no future trucks are put out on high-pressure tires, thereby initiating the moves which are necessary gradually to retire this type from the market.

Since the trend of the transportation set-up in our markets, with our excellent highways covering the entire country, is toward higher speed, lower air pressure in the tires is vital. While it hardly seems necessary, I think it is fair to remind the truck-bus industry that we could not possibly have passenger cars such as we now have, with their performance and operating characteristics as they are, on tires using the high pressure of the pre-1920 years; and it is just as logical to reason that solid tires could not possibly be used on our modern trucks; and that, for future truck and bus operations, the lower pressures are just as essential. While there is a temptation to write a long dissertation on this, I shall content myself with setting down the advantages of the balloon type of tire in topic form:

Balloon Bus-Truck versus High-Pressure Tires

Air-pressure recommendations average one-third lower for balloons than for high-pressure tires.

Benefits to Truck Operations.—(1) Saves maintenance on truck

(2) Protects fragile loads

(3) Permits faster average speed

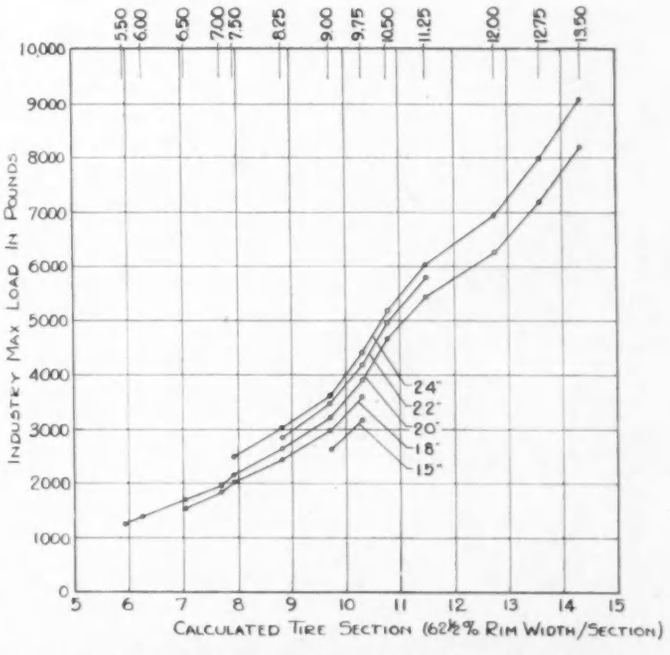


Fig. 18—Illustrating the Tire and Rim Association Maximum Load Tables Plotted Against Calculated Tire Sections on Rims 62½ Per Cent of the Tire Sections

Table 2—Lines of Tires for Buses

Present Balloon Line-Up							New Low-Pressure Line-Up					
Rated Maximum Load, Lb.	Nominal Size	Air Pressure, Lb. Per Sq. In.	No. Plies	O. D.	Rim	Center Spacing, In.	Nominal Size	Air Pressure, Lb. Per Sq. In.	No. Plies	O. D.	Rim	Center Spacing, In.
2,200	7.50–20	55	8	36.3	5.00S	10	9.00–18	40	6	36.2	6.00T	10¾
2,650	8.25–20	60	10	38.0	5.00S	10½	10.00–18	45	8	38.3	6.00T	11½
2,850	8.25–22	60	10	40.0	5.00S	10½	10.00–20	45	8	40.3	.....	....
3,250	9.00–20	65	10	39.6	6.00T	11½	11.00–18	45	8	40.5	7.33V	12¾
3,500	9.00–22	65	10	41.6	6.00T	11½	11.00–20	45	8	42.5	.....	....
3,900	9.75–20	70	12	40.7	6.00T	12	12.00–18	50	10	42.2	7.33V	13½
4,200	9.75–22	70	12	42.7	6.00T	12	12.00–20	50	10	44.2	.....	....
4,700	10.50–20	75	12	42.3	7.33V	13¼	13.00–18	50	10	44.7	8.37V	14¾
5,000	10.50–22	75	12	44.3	7.33V	13¼	13.00–20	50	10	46.7	.....	....

(4) Easier on driver
(5) Desirable on rough city pavements
(6) Empty trucks do not bounce as much
(7) Less tendency to sink into soft soil
(8) Better traction and braking control
(9) More desirable for trailers

Economic Aspects.—(1) Larger section; thinner walls flex easier and form less heat
(2) More tread rubber and less intensity of pressure
(3) Lower heating effect helpful to inner tube
(4) Cooler tire; less chance of failure from separation
(5) In dual tires, greater deflection equalizes the load better
(6) In general, will show lower cost per mile
(7) Lower pressure puts cords under less tension
(8) Larger cross-section gives less degree of flexing
(9) Loads carried per dollar are substantially the same
(10) Loads carried per pound of tire are substantially the same.

In trying to discover a way to show visually the superiority of the balloon-type tires over the high-pressure type for trucking service, I hit upon a plot, shown in Fig. 20, which

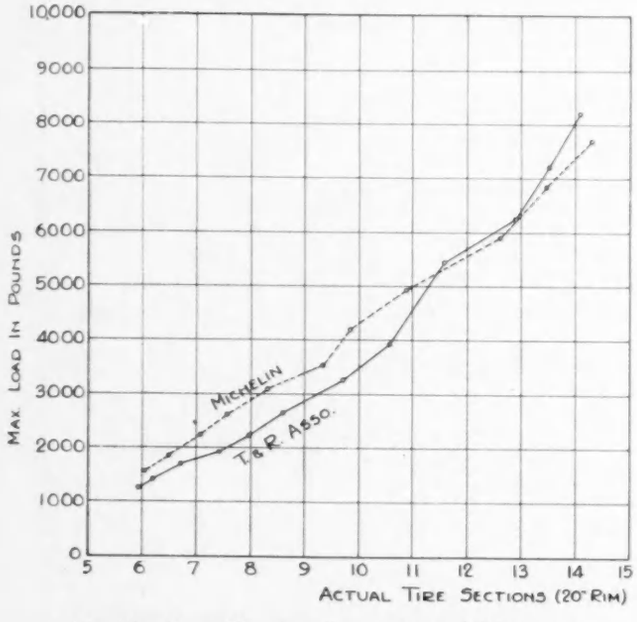


Fig. 19—Chart Showing Comparison of Maximum Load Ratings
Truck-bus balloon-tires—Michelin versus Tire and Rim Association

indicates a trend. An inspection of Fig. 20 will show quite clearly that the high-pressure tire is limited in high-speed application from the point of view of tire life, while of course it is known to be satisfactory for slow-speed work.

In the study of tire problems on buses, there is no problem of overloading for the simple reason that maximum loads on buses can easily be determined and the tire manufacturers handling this class of business have been able to convince the bus operators of the desirability of starting the buses out on the proper tire size. As public patronage of bus lines has increased during the last year or two, there has been an increasing amount of unfavorable comment on the riding characteristics of buses, all of which sums itself into a criticism of the air pressure in tires being too high. At the present time the company with which I am connected has in progress a new development in the shape of a new low-pressure line of bus tires. The pressures on this new line of tires are shown in the plot in Fig. 21, in comparison with the pressures of other prominent lines of tires. On this plot is one line labeled "Michelin Metric Balloon," included because such a line of tires has been offered by Michelin for several years for trucks. Table 2 also shows the characteristics of this new line of low-pressure bus-tires.

As the trucking business has expanded, the tire requirements have become more and more complicated. Not only do we have a large number of sizes of tires and two distinct lines of the high-pressure type, but we must also discriminate for different classes of service in our tread designs, and lately have found it desirable to recognize different haulage classifications. In a general way, I would emphasize the groups as follows:

(A) City, local and intercity trucking and bus lines
(B) Intercity bus operation
(C) City commercial deliveries
(D) Off the road, mud road, road maintenance, dirt-hauling jobs
(E) Strictly trailing vehicles in low-speed operations.

The tire equipment for Group *A* must be constructed to stand overloading and rough pavements, often in poor repair.

Group *B* tires must have a tread design which gives good traction in winter; and, particularly in hilly and mountainous districts, must be able to resist the detrimental effects of heat from long-distance high-speed runs.

The city commercial delivery group *(C)* needs primarily tread designs and volume of tread rubber to give the maximum possible wear in start-stop service, which very effectively scrubs off the tread rubber. In this group there is also the problem of discriminating between passenger-car tires and

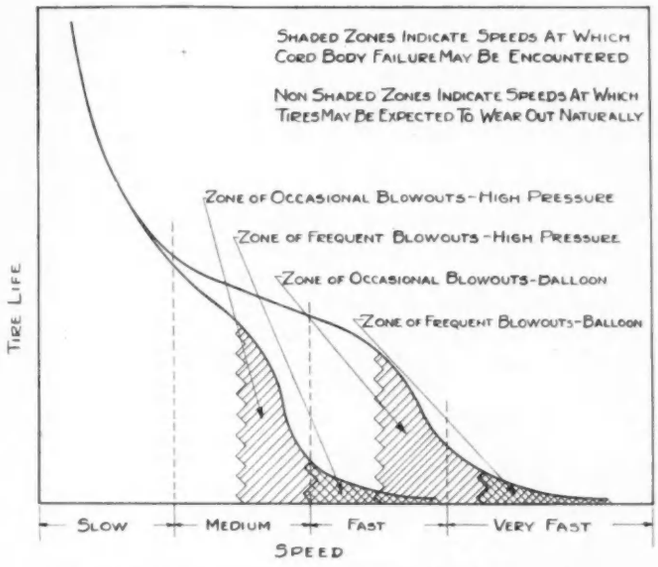

Shaded Zones Indicate Speeds At Which Cord Body Failure May Be Encountered
Non Shaded Zones Indicate Speeds At Which Tires May Be Expected To Wear Out Naturally
Zone of Occasional Blowouts-High Pressure
Zone of Frequent Blowouts-High Pressure
Zone of Occasional Blowouts-Balloon
Zone of Frequent Blowouts-Balloon
Tire Life
Slow
Medium
Fast
Very Fast
Speed

Fig. 20—Visualization of the Superiority of the Balloon-Type Tires Over the High-Pressure Type for Trucking Service, Indicating a Trend

truck tires. For ½-ton commercial-jobs, panel delivery and open body, the passenger-car tires on passenger-car drop-center rims unquestionably are the most economical and give the best all-round satisfaction. Probably for the ¾-ton-capacity jobs, the same reasoning would apply. When we get to the 1-ton jobs, it is the tire companies' contention that this is a debatable point. If the 1-ton job under consideration is strictly a commercial-delivery job, the passenger-car tires, even in dual combination on drop-center rims, are satisfactory, say in panel delivery service; but if this job is worked out in such a way that it is really a truck subject to overloading and abuse, to which small trucks are notoriously subjected, then the tire companies insist that truck tires of a suitable size on truck rims should be used.

In the next group *(D)* the load can be provided for by adequate tire-section size, but some form of a super-traction tread is necessary and load-carrying tires with large section, few plies and low pressures are necessary.

On trailers and semi-trailers, Group *(E)*, which are often used in intercity work, the problem is mainly one of tread wear.

Another interesting point to bring up is some new thoughts developed in the last two or three years on the manner in which operating conditions offer a different basis for deciding load ratings. Experience is teaching us that speed and traction versus trailing wheels make a lot of difference. We feel that, technically, as the speed limits go down, we can allow much more load to be carried. As a way of conveying the idea, we could set up increasing load allowances on some such basis as the following:

Speed, m.p.h.	Rating, Per Cent
35 and up	100
21 to 35	120
8 to 20	135
Up to 8	160

As a matter of fact, some of our tables already show differentiation along this line; but the commercial handling of such a step is so difficult to control that at the present time we do not see our way clear to make such a complicated move. Neither do we see how we could commercially control an authorization for trailer tires to carry heavier loads than traction-wheel tires.

Before closing this paper, it seems appropriate to emphasize certain aspects of the application and use of tires which should be observed by truck manufacturers and truck operators to get the best out of tires.

In selling trucks, it should be the salesman's duty to ascertain full information about the use to which the truck is to be put by the operator; loads, type of material hauled, roads over which it will run, facilities for taking care of the tires, and the like. A tire of the proper size and tread design to give maximum satisfaction should then be recommended from the options provided. The proper rim and center spacing is equally important. The type and size of tire having been selected, the rim and center spacing equipment should be those shown on Page 36 of the 1935 Tire and Rim Association Handbook under the heading "Original Equipment Recommendation". By so choosing, the rim and center spacing will be adequate in the event of later discovering that a larger-size tire is necessary so that a "consumer-oversize" tire can be put on without any expensive wheel or rim change.

The effect on front-tire wear of improper wheel alignment in front-axle parts is a factor which is extremely important. With the correct alignment, it is fair to say that front tires ordinarily should deliver in the order of twice as much mileage as the rear tires, but if the alignment is incorrect in some particular, a very serious loss of tread wear may result. Ordinarily, truck drivers do not pay much attention to the condition of the front tires; while on the other hand intercity bus operators, because of the higher speeds at which their schedules are set, are very much "safety conscious" of the front end and put only the new tires on front wheels. I want to register at this point my belief that front-end design should provide for a camber which comes between ½ and ¾ deg. per wheel. A higher camber is definitely responsible for excessive front-tire wear and particularly uneven "tread

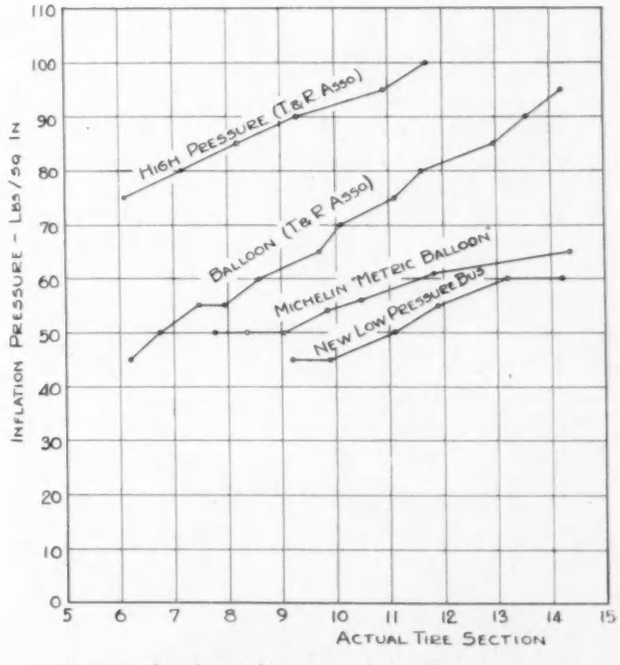


Fig. 21—Graphical Showing of the Relation of the Prominent Inflation-Pressure Tables

wiping" or "heel-and-toe action", as it is sometimes called. The ideal condition would be to have the wheels straight up and down, but because of the danger that, as the parts wear or fatigue, negative camber will be developed, it is found desirable to make the ½ to ¾-deg. recommendation.

While it may be rather a hackneyed subject to some designers, I must not overlook the opportunity to emphasize again the need of designing in such a way as to eliminate brake heat against the beads of the tires and provide ventilation in the wheel housing. Particularly on buses, we see a definite tendency to go to smaller-diameter rims. I feel that our next step on intercity buses is to go to 18-in. wheels, and, of course, this is going to bring in a pretty serious problem of how to get adequate brakes. It will have to be done, and proper measures will have to be adopted to be sure heat does not reach the beads of the tires. The up-to-date body-designs are tending toward housing-in the tires in a way that interferes with good ventilation. This housing-in must not be done unless some effective ventilation is provided for otherwise. In this case it is not a matter of heating the beads, but raising the temperature of the tire as a whole, and particularly on buses operating in long-distance runs at high-speed schedules in hot parts of the country.

There are some aspects about the use of dual tires which ought to be brought out and emphasized. The problem here subdivides into two phases: first, the dual tires on the driving axle on the rear of a truck or truck-tractor; and second, the dual tires on a semi-trailer or full trailer. It is quite common to observe "funny" wear, either on the inside or outside tires. There are many factors which influence this, such as the load, inflation, speed, crown of the road, mating of tires, and the like. From a vehicle-design point of view, however, the strength of the axle is very important. This has been so impressive to us that we have taken the trouble to measure the load distribution on four tires on certain axles. Because of axle deflection, the inside tires on the driving wheels of a truck-tractor carried the overload and showed what we call "overload-wear"; that is, the center of the tread stood up much higher than both shoulders. The outside tires wore smooth at a more rapid rate because of the fact that the inside tires, being deflected more, controlled the angle of velocity of rotation, which obviously produced the scuffing effect over the complete surface of the outside tire. This job was weighed after it had delivered a total of 14,000 miles, the loadometer scales being used in such a way that we actually obtained a correct picture of the load per tire. The figures were as follows:

	9.75-20 *Tires*		
	Weights, Lb.		*Per Cent*
Position	*Empty*	*Loaded*	*of Load*
Right Rear Outside	1,680	3,870	20.5
Right Rear Inside	2,300	4,900	26.4
Left Rear Outside	1,850	4,640	24.9
Left Rear Inside	2,500	5,250	28.2
Total	8,330	18,660	100.0

The same unit was weighed later after it had gone 28,000 miles and the weight figures show the following:

Right Rear Outside	1,400	4,050	20.25
Right Rear Inside	2,750	5,750	28.75
Left Rear Outside	1,500	4,350	21.75
Left Rear Inside	2,600	5,850	29.25
Total	8,250	20,000	100.00

The "funny" wear in dual trailer-tires when the axle sags is somewhat different. We have observed an extremely irregular and uneven wear over both surfaces of both the inside and outside tires. A common practice of truckers is to have their trailer axles "arched" periodically at some local service station in an effort to avoid this uneven tire wear. While I am familiar with the fact that in the last two or three years steps have been taken to furnish stiffer axles of improved design, we are of the opinion that, in some instances, there is still too much overload and too much axle bending. We have even seen a few cases of trailer axles bent in a fore-and-aft direction which produced a toe-in effect with its consequent unfavorable tread wear.

Only recently we had occasion to check the performance of tires on a semi-trailer that came to our shipping dock, and the results are so interesting that I give below the weight figures which show that the axle must have had a peculiar twist to give the results indicated:

Position	*Loaded Weights, Lb.*	*Per Cent of Load*
Right Rear Outside	4,950	23.1
Right Rear Inside	5,950	27.7
Left Rear Outside	5,800	27.0
Left Rear Inside	4,750	22.2
Total	21,450	100.0

Apparently, this axle was "arched" on one end and "sagged" on the other. We also had photographs taken of the tires as they were actually installed on the trailer, as shown in Figs. 22 and 23.

The problem of insuring safety against front-tire blow-outs in high-speed bus-operation is uppermost in the minds of operators. The present practice is to use only new tires on front wheels and remove them when they have seen only a part of their service. As far as I am informed nobody, up to the present time, has been able to work out any safety treatment to supplement the tire in case of a blow-out; that is, one which is practical in every way and sufficiently reliable to give 100 per cent protection. I feel personally that bus designers should seriously undertake to design dual front equipment to provide this added safety. While it brings in complications in steering, braking, and also in making each wheel rotate independently in dual combination, in order to steer easily, as soon as somebody tackles this job and works it out, that much sooner will a very important phase of bus design be over with.

The important things for the operator to practice and observe are:

(1) The inflation of the tires should be checked frequently. As mentioned earlier in the paper, it is the air which carries the chief part of the load, and if there is not sufficient air in the tire, a definite loss of tire mileage will result. Tires should not be over-inflated and neither should truck drivers "bleed the pressure". The pressure recommendations as made by the tire companies presuppose a cold tire.

(2) In the case of duals, tires of the same state of wear should be mated together, and in the event of different makes of tires having to go on the same wheel under dual combination, pains should be taken to see that tires are selected which are reasonably close to the same overall diameter.

(3) Reasonably frequent check on wheel alignment is very important. Owners should be made to realize that in spite of rugged construction and careful workmanship, front wheels

are likely to get out of alignment and a great deal of tire mileage is lost if alignment inaccuracies are not corrected.

A word about repairs and retreads. It generally works out in the operation of any fleet that retreading becomes important because of the need of added non-skid mileage where otherwise the tires would be running on smooth rubber. Regrooving has been resorted to, to some extent, but this is really only a makeshift. During the last few years a good deal of attention has been paid to developing improved methods and technique in perfecting much more satisfactory retreaders. It is this situation that has brought about the wide distribution of modern full-circle retreading-equipment which will vulcanize a tire tread satisfactorily. With a recognized first-line tire, proper inspection, workmanship and materials, satisfactory mileage will be obtained on retreaded tires with an actual "cost per mile" decrease.

Generally speaking, it is best to select tires for retreading before they have worn through to the fabric, because then the tire is likely to be relatively free of cuts and local separations. No tire should be retreaded that has been badly cut through and repaired, or which shows upon inspection any evidence of ply separation or fabric fatigue, such as beaking down of the cords in the shoulder area of the tire at the edge of the tread design. Of course, to insure a perfect inspection and resultant good retread, it is necessary to remove all of the old tread, cushion and breaker and replace these with all-new material. Some shops "recap", which simply means roughening the old tread and adding new tread-rubber. This method requires expert judgment, and a more careful selection of tires to insure success, because small separated areas that may exist underneath the tread cannot be detected and will later cause tire failure.

It is difficult to make a general statement on the mileage that can be obtained on retreads because of the varying conditions of the tires at time of retreading and varying operating conditions. Our experience indicates that from 15 to 25 per cent of the tires of a fleet can be retreaded, and that the net average mileage increase on a fleet basis should be from 10 to 20 per cent. It generally works out that the retreaded group of tires on any fleet will run from 70 to 80 per cent of the original tire mileage.

Repairs on tires have been made successfully for a long period of time; but, as in retreading, tires must be carefully inspected before a repair is made to be sure the tire justifies the repair cost. Injuries that are more or less snags or external cuts should be promptly vulcanized and reinforced on the inside where necessary to prevent spreading of the injury. Tires that are cut through require special attention; and, generally speaking, an injury of over 4 in. is seldom worth repairing. Here again, however, operating conditions and the use to which the tire is to be put after repair govern what can be done.

We are often asked the question "Should retreads and repairs be 'babied' when they have been put back into service?" Work of this kind is dependent on getting proper adhesions by buffing, cementing and curing new material to material which has already been vulcanized and has been in service. While our processes for accomplishing this have been worked out to a very satisfactory degree, obviously, the adhesion of the new work to the old cannot be as nearly perfect as that of the original plies of the tire, which was cured all the way through as a unit. Therefore, we find it is reasonable to take the position that repairs and retreads executed with up-to-date equipment, up-to-date methods and using fresh materials, can be expected to wear out without giving trouble under average operating conditions. If possible, it is always ad-

Figs. 22 and 23—Illustrations of the Effects on Dual Trailer-Tires Installed on an Axle That Was "Arched" on One End and "Sagged" on the Other

visable to put retreaded or repaired tires on the truck in what might be called the easiest position, such as the outside rear in duals, or, if trailers or semi-trailers are used, where they do not have to transmit power. Practically the only type of operation where we have found it not advisable to use retreads and tires which have major repairs is in the very fastest intercity service.

Puncture-proof tubes are available for markets where puncture hazards are excessive, such as desert country where thorns get into the tires and are hard to combat. In our experience puncture-proof tubes have been very helpful in eliminating road delays on local hauling jobs and in start-and-stop service, and can be recommended for such use. In high-speed service, where operations are over good roads, we do not advise their use, because of the heat that develops in the tire due to the thickness of the puncture-proof-type tube, and the resultant bad effect on tire life. In our opinion there is no satisfactory way to treat an ordinary inner tube to make it puncture proof.

In conclusion, I might simply make the observation that reputable tire manufacturers now have sufficient control over their manufacturing operations, materials used and the curing of the tires and tubes, so that such things as imperfect workmanship, materials and manufacturing, are very infrequently encountered. We feel that the mileage is built into the tires and that the operators and owners can get the mileage out of the tires if they are used properly. It is up to the tire manufacturers and truck manufacturers jointly to educate the users in buying original trucks properly tired and caring for tires after they have come into their possession so effectively that they will get the proper economy and satisfaction.

Discussion

Major Change in Load Ratings Inadvisable

—Sidney M. Cadwell and A. W. Bull

United States Rubber Products, Inc.

MR. HALE has presented an interesting paper which touches on many of the factors involved in truck-tire load-ratings. We agree with his position that any major change in load ratings is inadvisable, and we wish to present additional evidence to support this view.

The fundamental question: "Should the standard loads be increased?" has been examined by considering the possible effects on the groups involved. Increasing the rated loads will give the truck manufacturer a slightly lower initial cost which he may or may not pass on to the truck buyer. In return, he will get a decided increase in complaints and dissatisfaction with his products if it develops that they are under-tired.

To the operator, the change will mean, in some cases, a lower first-cost, and a lower cost per tire for replacement. However, replacements will be made sufficiently oftener to more than offset the difference in unit cost; operating costs per ton mile will increase; road delays due to tire failure will certainly be more frequent; and the ability to overload in emergencies will be reduced.

To the tire manufacturer, the change will mean an increase in adjustment costs and a lot of dissatisfaction with the product. The tire maker in turn can only defend his position by suggesting oversizing, which immediately reflects the fact that the vehicle was under-tired when delivered by the manufacturer.

The truck manufacturer has one additional advantage from such a change. Tires already act to a large extent as safety valves by preventing too great an overload. By reducing tire size he will obtain still greater protection in this respect, but it is certainly questionable whether it would be good business for him to take advantage of this situation.

Irrespective of any agreement on tire loads, we must face the fact that actual loads are determined by what the traffic will bear. In general, truck owners know to what degree they can overload a tire, and they will put on all the load that can be economically carried, taking into account overall costs, the frequency of tire failures, and the leniency of their particular tire adjusters.

The best proof of this statement is the fact that, although great improvements in truck tires have been made in the last seven years, the percentage of adjustments has remained quite stationary. This is shown in Fig. *A*. The records are those of the Rubber Manufacturers' Association and are for adjustments by all the larger companies and most of the smaller ones. The percentage of adjustments for truck and bus tires has remained fairly constant at a level approximately four times higher than passenger-car adjustments during the same period.

The distribution of adjustments by sizes is shown in Fig. *B*. These again are data for 1934, from the Rubber Manufacturers' Association. All sizes are worse than passenger tires and the percentage of adjustments increases sharply as tire

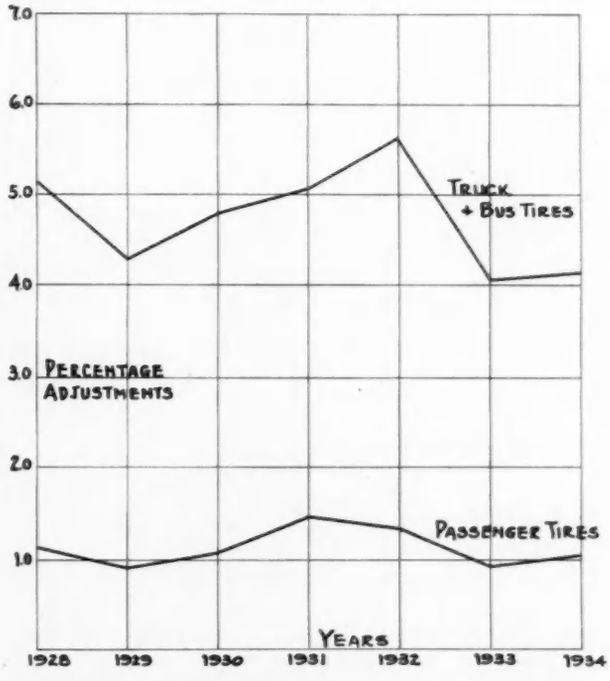


Fig. *A*—(Cadwell-Bull Discussion) Percentage Adjustments of Tires by Years. Data from the Rubber Manufacturers' Association. Only the First Quarter of the Year 1935

Year	Passenger Car	Truck and Bus	Year	Passenger Car	Truck and Bus
1928	1.15	5.17	1932	1.33	5.64
1929	0.92	4.28	1933	0.92	4.07
1930	1.09	4.79	1934	1.05	4.14
1931	1.47	5.06	1935	1.64	4.01

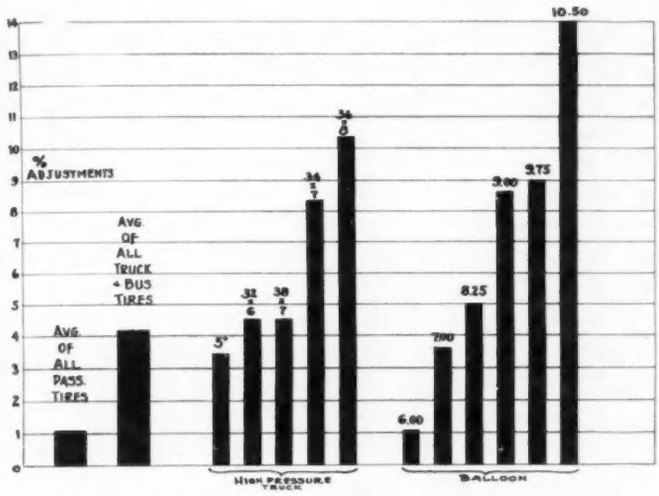


Fig. *B*—(Cadwell-Bull Discussion) Adjustment Records for 1934 of the Rubber Manufacturers' Association

size is increased. This probably means that our standard loads on the larger sizes are already too high and should be reduced.

Of course, it may be argued that these larger tires are more consistently overloaded than the smaller tires, but the facts directly contradict such an opinion. The Bureau of Public Roads, the Johns Hopkins University, and the State of Maryland have recently published some cooperative research in *Public Roads* for May, 1935, page 37, in which they submit

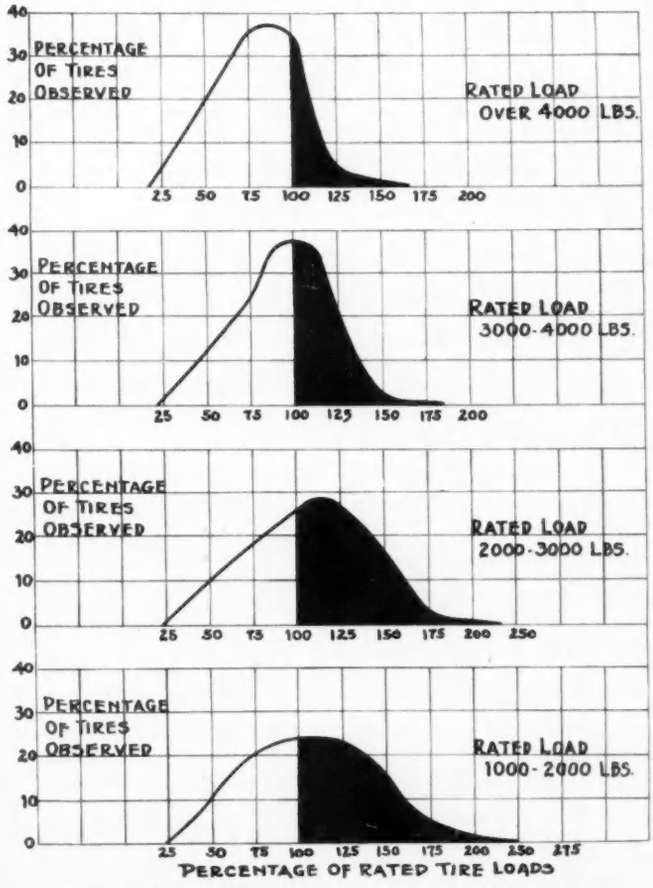

PERCENTAGE OF TIRES OBSERVED
RATED LOAD OVER 4000 LBS.
RATED LOAD 3000-4000 LBS.
RATED LOAD 2000-3000 LBS.
RATED LOAD 1000-2000 LBS.
25 50 75 100 125 150 175 200 250 275
PERCENTAGE OF RATED TIRE LOADS

Fig. C—(Cadwell-Bull Discussion) Distribution of Actual Loads Compared with Rated Loads

data on actual vehicle weights for more than 10,000 vehicles traversing the highways near Baltimore in the summer of 1934. Their records include the actual weights on each axle, the tire sizes, and the ratios of actual tire loads to rated tire loads.

Fig. *C* shows the distribution of actual loads compared with rated loads. The shaded sections represent overloads, and the frequency of overload is greater with small tires than with large ones. More than half of the tires in the two lower curves are overloaded from 25 to 50 per cent, and some were overloaded more than 100 per cent. These distribution curves are plotted from data in Table 21, page 46 of the *Public Roads* article. It has been assumed that the exact ratios given in the table were averages of a range of loads. Thus, where the table shows 22.8 per cent of the tires in the 1000-2000-lb. class as carrying 125 per cent of rated load, it is assumed this covers the range 112.5-137.5, which averages 125 per cent.

This factual evidence from an unbiased survey makes it easy to understand why heavy service-adjustments are out of all proportion to passenger-car adjustments. We would also like to call attention to the fact that these values are derived by assuming that the loads are borne equally by dual tires. Actually, this is very seldom the case, inside duals nearly always carrying more of the load than the outside tires. If this is taken into consideration, the overloading of individual tires is worse than shown in the survey.

Another factor in tire performance which has been quite completely ignored is the matter of speed. Speed has just as much to do with tire performance as the load carried. Truck speeds have been pushed steadily upward and this additional burden on the tires has been met by the tire maker sufficiently well to keep the adjustments nearly constant. We have made a number of tests in which speed and load have been treated as independent variables, and they show that carcass failures are roughly proportional to the product of load and speed. If our present tire is satisfactory at 40 m.p.h. with rated load, it will give about the same service at 30 m.p.h. with a 33 per cent overload; and at 50 m.p.h. the load for equivalent service must be reduced to 80 per cent of the standard. On slow-moving vehicles an increase in loads might be permitted, but it should be balanced by a reduction in present standards at high speeds.

In view of all of the facts which have been presented, a revision of the standard loads upward is unreasonable, except for slow-moving vehicles actually incapable of high speeds and not merely on low-speed work temporarily. The tire industry has been burdened with excessive adjustment costs in heavy-service tires for years, and it can see no good reason for increasing this burden. Raising present load limits will merely give dissatisfied users additional grounds for argument.

To say that, because overloading is common, we should raise the ratings to make the standards fit the practice, is as illogical as to argue that where reckless driving is common we should modify traffic laws to permit it.

New Method for Determining Temperature Inside a Tire

—K. D. Smith

B. F. Goodrich Rubber Co.

MR. HALE has covered the subject so thoroughly that there can be little doubt as to when a truck tire is overloaded. Neither should there be any question regarding

the present load and inflation tables as established by the Tire and Rim Association.

We have made some independent observations regarding the effect of speed and load on tire life, and I wish to present the results of this work merely as a supplement to Mr. Hale's paper. It has been my firm belief for some time that the Tire and Rim Association must augment the present tables, taking into consideration the effect of speed as affecting temperature. You may be sure this feature is being given consideration at present by the individual members of the Association, and it is hoped that we can formulate a definite program to this end in the near future.

Since we know that the flexing life of a cord and its ability to adhere to rubber varies with the temperature at which it is flexing, it seems foolish to disregard a tire's carcass-temperature when attempting to arrive at its load-carrying capacity. We have recently developed a technique for the taking of exact temperatures within a tire at any desired point which we find is infallible and under controlled conditions can be duplicated time and again. The controlled conditions consist of running the tire on an indoor driven wheel enclosed in a room with controlled temperature. This indoor machine is similar to the one used by the National Bureau of Standards and permits applying definite loads. Space does not allow me to go into details on the method of taking the temperatures, but Figs. *A* to *E* show data obtained by this indoor testing which, in a way, are similar to those given by Mr. Hale.

Fig. *A* shows the effect of load on the carcass temperature with a constant speed and atmospheric temperature. Fig. *B* shows the effect of speed on temperature in the carcass with load and atmospheric temperature constant. Fig. *C* shows the effect of change in atmospheric temperature on the carcass temperature with speed and load constant. Fig. *D* shows speed plotted against temperature with various curves of different loads. Fig. *E* presents the same data as in Fig. *D*, shown in a different manner but giving a clear indication that loads must be decreased when speeds are increased. These data clearly indicate that the temperature at which a tire operates, which may be affected by speed as well as load, must determine the carrying capacity or life that may be expected.

Since pressures and temperatures as well as loads are governing factors of tire performance, we wish to list several rules that may be used as a guide toward better truck and bus tire service, as follows:

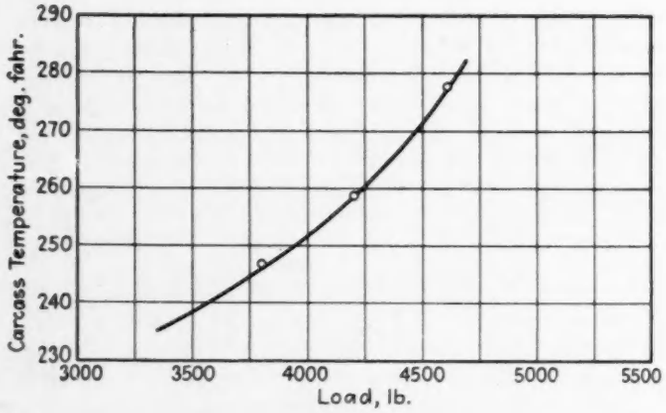


Fig. *A*—(K. D. Smith Discussion) Load versus Carcass Temperature; 10.50-22 Tire

Constant speed, 45 m.p.h. Atmospheric temperature, 115 deg. fahr.

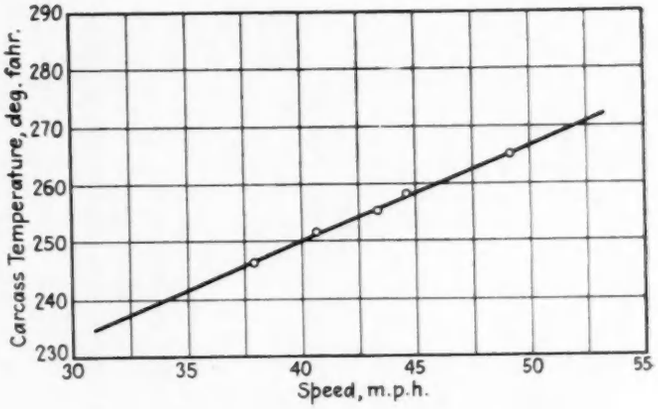


Fig. *B*—(K. D. Smith Discussion) Speed versus Carcass Temperature; 10.50-22 Tire

Constant load 4200 lb. Atmospheric temperature, 115 deg. fahr. A 1-m.p.h. change=1.6 deg. fahr. tire temperature

(1) Pressure increases should never exceed 25 to 30 lb. Operating conditions that cause tire pressures to increase more than 25 to 30 lb. are too severe and cause the internal temperatures to be too high, and the high pressures developed reduce the tire's resistance to bruising.

(2) We do not recommend bleeding to reduce this pressure, as this only increases the tire's running temperature because of the increased deflection.

(3) Rule (2) can be readily understood when we realize that a tire's pressure will stop increasing only when the temperature developed through the tire's deflection equals the temperature carried off by radiating to the atmosphere. Since the amount of temperature developed decreases as the deflection decreases and the amount of temperature radiating to the atmosphere increases as the temperature of the tire's surface increases, the point at which these two coincide is the point where the pressure no longer rises. By reducing this pressure, we immediately upset the equilibrium by increasing the amount of heat developed because of the increased deflection. This equilibrium will, therefore, be reached again only when the temperature of the tire becomes sufficiently higher to make up for this increase in the temperature input.

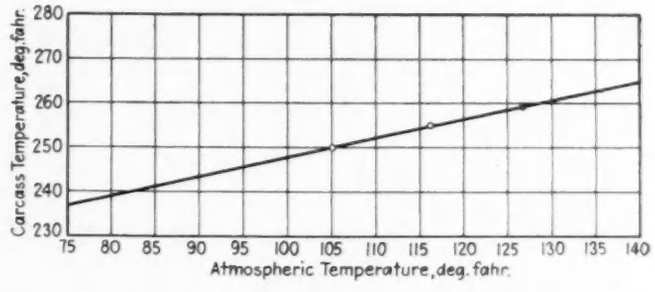


Fig. *C*—(K. D. Smith Discussion) Atmospheric Temperature versus Carcass Temperature; 10.50-22 Tire

Constant load, 4200 lb. Constant speed, 45 m.p.h. A 1-deg. room-temperature change=0.45 deg. fahr. tire temperature.

(4) We recommend a pressure differential for dual tires ranging from 5 to 15 lb., usually making the outside tire one-half this differential higher than rated pressure and the inside tire one-half the differential lower than rated pressure. We do this because the inside tire always increases in pressure higher than the outside, because it does not have the ventilation and is also directly over the brake drum. Since this inside tire pressure does increase to a higher figure than the outside, it carries greater than 50 per cent of the load. We therefore give it this pressure differential so that when the two tires reach the point of equilibrium they will both have approximately the same pressure, and, consequently, will both be carrying 50 per cent of the load. On highly crowned roads, it is usually found necessary to use a higher pressure differential than on relatively level roads.

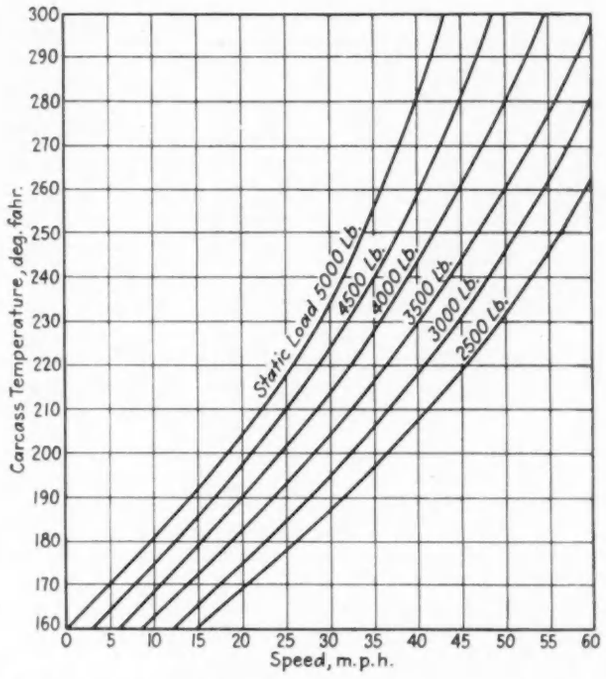


Fig. *D*—(K. D. Smith Discussion) Speed versus Carcass Temperature; 10.50-22 Tire

With constant load and at a constant atmospheric temperature of 115 deg. fahr. The initial conditions of the air in the tube were: Air pressure, 75 lb. per sq. in.; air temperature, 80 deg. fahr. The carcass data shown are from the point of maximum temperature

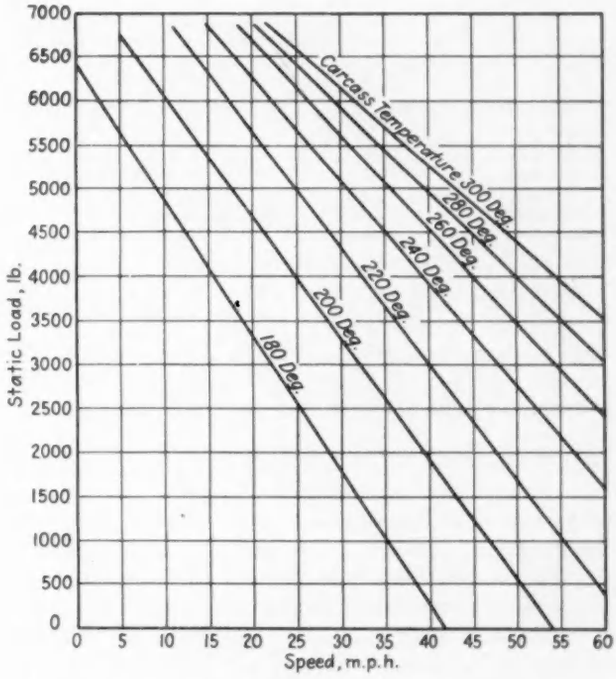


Fig. *E*—(K. D. Smith Discussion) Load versus Speed; 10.50-22 Tire. Constant Carcass and Atmospheric Temperatures

At a constant atmospheric temperature of 115 deg. fahr. The initial conditions of the air in the tube were: Air pressure, 75 lb. per sq. in.; air temperature, 80 deg. fahr. The carcass data shown are from the point of maximum temperature

Tire-Mileage Expectancy Versus Tire Load Analyzed

—**Leo Huff**
The Pure Oil Co.

A CERTAIN amount of reticence is felt about voicing any ideas contrary to the statements made in Mr. Hale's paper, principally because the statistics presented are undoubtedly taken from well-grounded experiments and tests which have been conducted by capable tire engineers, and also that contrary comments may be made without full appreciation of other fundamental controlling factors involved which are not mentioned.

Under these circumstances, any opinions or deduction resulting from experience which cannot be reconciled with the chart presented by Mr. Hale, in which he relates tire-mileage expectancy against tire load, are made with the sincere intent of adding clearness to this problem. According to this chart the lighter the load on a certain size of tire the greater the mileage expectancy, and the mileage expectancy increases at a much more rapid rate than the tire load is decreased. This chart also indicates that any increase in tire load above normal results in a decrease in tire expectancy, but that tire life decreases at a much more rapid rate than the load increases. This chart indicates that overload is disastrous to tires, which my experience very satisfactorily bears out; however, I cannot agree that any appreciable decrease in load below the normal capacity of the tire results in an increase in tire life in the proportion shown. Actually, I have found by experience that decrease in load below normal results in an increase in tire life only up to a certain point, but beyond that the trend of affairs reverses and a further decrease in load results in extremely fast tread wear.

In the chart, there is no mention made of inflation pressure. Assuming that the tire pressures are kept rationally in proportion to the tire load I find that, under certain circumstances where the tires were abnormally underloaded, any reduction of air pressure to keep the tire soft enough to stay on the ground was not adequate to provide proper wear.

From the viewpoint of the average operator, cost is paramount. Depending upon the nature of the analysis, costs are either figured on a per-mile or a per-ton-mile basis, but if the chart as presented by Mr. Hale is extended to a cost-per-ton-mile basis for underloaded and overloaded tires, we find the chart indicates that it is decidedly advantageous to oversize or to apply a larger tire where the loads to be carried exceed the normal load of the tire that is on the vehicle.

For example, if we arbitrarily choose a tire size having a certain capacity, and costing so much on a certain base price and operate this tire at 120 per cent of normal load, we find the cost per ton mile to be approximately one and six-tenths ($0.0016) mills; however, if we replace this tire with the next largest tire size, fitting the same rim, we find that with the same load that would overload the original tire, our cost per ton mile according to Mr. Hale's chart, is roughly one and three-tenths ($0.0013) mills. I would say that this would be in line with my experiences provided operating conditions were within normal operating limits. I have had, however, some very disastrous results with overloaded tires, which, without a question of doubt, would indicate that overload increases the cost per ton mile at a still much more rapid rate than Mr. Hale's chart would indicate, but I know that the circumstances under which this occurred were abnormal since the pavements were anything but good and the results were not due primarily to the dead weight on the tires but to the extraordinary hammering and battering that the tires

Table *A* (Huff Discussion)—Extension of Mr. Hale's Chart on Tire Life Versus Percentage of Load

9.75–20 Tire					10.50-20 Tire	
Normal Life...35,000 Miles Cost.....$90.70 Capacity .....3,900 Lb.					Normal Life...35,000 Miles Cost.....$104.95 Capacity .....4,700 Lb.	
Normal Load, Per Cent	Load, Lb.	Life, Per Cent	Miles	Cost Per Ton Mile	Load, Lb.	Cost Per Ton Mile
70	2,730	200	70,000	$0.00095	3,290	$0.00091
80	3,120	160	56,000	0.00104	3,760	0.00100
90	3,520	127	44,400	0.00116	4,230	0.00113
100	3,900	100	35,000	0.00133	4,700	0.00128
110	4,300	82	28,700	0.00147	5,160	0.00142
120	4,680	68	23,800	0.001625	5,650	0.00156
130	5,080	57	19,800	0.001805	6,100	0.00174
140	5,460	47	16,400	0.002030	6,580	0.00195
150	5,850	39	13,600	0.002280	7,050	0.00219

received in service without having their normal quota of safety factor to protect them.

Although, as outlined in the foregoing, it is agreed that overloading is extremely expensive and that operators should concern themselves with the evil effects of overloading truck tires, we believe there is still something to be said about the underloading of tires. I have already pointed out that my experience with underloading does not conform to the relationship shown on the chart but that, even giving due consideration to an adjustment of air pressure to agree with the reduction in load, bearing in mind that the majority of front-wheel tires are very much underloaded, I have found, in a great many cases, that it is exceedingly economical to use smaller tires on these front wheels than on the rear. Circumstances alter cases, and the answer to whether or not different size tires should be used front and rear depends largely upon other relevant circumstances.

Having previously reviewed the tire-manufacturers' viewpoint with respect to overloading and underloading of truck tires, and attempting to conciliate them with the experiences of a fleet operator, it seems difficult from a tire-operators' standpoint to attach too much significance to an established load-inflation rating of a truck tire. Due to service conditions tires which are overloaded fail prematurely, while, on the other hand, overloaded tires produce mileages far above the average; then how can one be easily convinced that a dead-load rating for tires is the logical basis? Attaching too much significance to a normal load-inflation rating of tires would therefore seem to be disadvantageous to both the tire manufacturer and operator.

The thought comes to mind that perhaps the reason why overload abuse is so prevalent today and why so many operators have serious truck-tire difficulties is because these operators are basing their choice of tire sizes on standard load-inflation schedules, and, knowing that other operators get good results when running tires overloaded, they feel there should be no reason why they should not run their tires overloaded and expect the same results. Obviously, the answer lies in something beyond the dead weight on the tires, and many operators find themselves in a position where they are getting unsatisfactory service and high tire costs by predicating their choice of tire equipment purely on a dead-load basis instead of providing sufficient tire capacity on a shock-load basis.

It is true the dead-load basis of rating tires is tangible and the shock-load basis is very abstract; however, when actual experience proves tires have insufficient or surplus capacity for the load-service they are called upon to perform, then it would seem reasonable to throw overboard our excessive faith in rating tires purely on a dead-load basis and apply tires of such size as may prove adequate and provide the most reasonable cost per ton mile.

There is no question but that the quality of truck tires by and large has been considerably improved, and we have every reason to expect that there will be similar improvements and developments in the future. However, it seems that the results obtained from truck tires today vary over much wider limits than ever before. The logical explanation of this is that the increased power of vehicles, the higher available speeds, more congested traffic conditions and more effective brakes, shorten the life of tires in greater proportion than the tire manufacturer can improve their quality. On the other hand, we still have tires of better quality operating under conservative service conditions under which the average mileage will reach a ridiculously high figure.

Mr. Hale's comments on the servicing and care of tires are extremely interesting. Aside from the choice of the proper size and type of tire fleet operators are learning that a reasonable amount of care and attention pays big dividends in tire savings. From my point of view, tire manufacturers are attempting to take the hocus pocus out of their instructions for the proper means of obtaining the most satisfactory performance and longest life and, having boiled it down to the simple understandable process of consistent, accurate inflation and the proper mating of tires, have, I believe, benefited the fleet operator more than all the reams of uninteresting and hard-to-understand literature which has been broadcast in the past and which meant so little.

Table *A* was prepared as an extension of Mr. Hale's chart on tire life versus percentage of load. In other words, based on a 9.75-20 tire at a base cost of $90.70 for tire and tube and a capacity of 3900 lb., the chart shows that, at 100 per cent normal load, the tire would carry 3900 lb.; and at an estimated base average life of 35,000 miles, would cost $0.00133 per ton mile to operate under these circumstances. Above and below these figures are shown the relative cost per ton mile for the percentages of underload and overload as shown. Against this is compared a 10.50-20 tire with a normal life of 35,000 miles at a normal capacity of 4700 lb. and cost $104.95. At 100 per cent load, which is 4700 lb., the cost per ton mile would be $0.00128. Comparing this 4700-lb. load on a 9.75-20 tire, which is approximately 120 per cent overload, the cost would be $0.001625, indicating that, according to this chart,

it is much more economical to oversize the tires on a vehicle when it is overloaded.

In closing, I would like to say that I feel very much encouraged by the manner in which the tire manufacturer and the fleet operator are getting together for the benefit of all parties concerned, and emphasize that it is hoped any statements or comments contained in this brief are made with the intent of adding to the general store of knowledge and experience which is now available for the manufacturer on the performance of truck tires. If any of these statements are contrary to the results of accurately conducted tests, to the experience of other operators, or to the general knowledge now had by the tire manufacturer, it is logical that there is still a good reason why these experiences are contrary, and it is to the benefit of everyone concerned that such differences be studied and logically explained.

Objection Made to Wider Cross-Section Tires

—L. V. Newton

Byllesby Engineering and Management Corp.

I DESIRE to compliment Mr. Hale upon his excellent paper, but I am somewhat disappointed in that he does not tell us the answer to the question: "When Is a Truck Tire Overloaded?" I conclude that so many variables are involved, that a direct answer is impossible.

The question of tire ratings is very important to the operator in that the gross tire rating set by the Tire and Rim Association determines the gross allowable vehicle and load weight.

In purchasing trucks from the manufacturers, we always determine from them their recommendations as to tire size and never do we place larger tires on a given truck than is recommended by the manufacturer of the vehicle.

If the Tire and Rim Association ratings are 10 per cent lower on tires sold in this country than upon the same tires sold abroad, then we can only conclude that the ratings are conservative here and less conservative abroad, or vice versa. I feel that an international rating should obtain.

I agree that high-pressure pneumatic-tires are no longer desired on new vehicles. They must, however, continue to be manufactured for some time to come for those trucks and buses now so equipped.

I believe intermediate balloon tires are needed, the spread between the various sizes now obtainable being too great.

I cannot subscribe to the use of larger cross-section lower-air-pressure tires on trucks with a rim diameter of 18 in., as has been suggested. Overall width of trucks is pretty well established at 96 in. Wider cross-section tires will certainly mean narrower frames and wider axles. This will interfere with our standard body measurements; in fact, it will destroy the standardization now in effect. I do not believe the advantages of lower-air-pressure tires are worth the loss of body-dimension standards and the expense to which the truck and axle manufacturers will be put.

I am disappointed that time did not permit Mr. Hale to tell us more about the retreading of tires. We have found that, when tires are wisely selected for retreading, we obtain about 80 per cent of the original mileage from the retread at about 50 per cent of the casing cost.

In closing, I wish to state that, as operators of a large number of motor vehicles, we are well pleased with the performance of the tire of today. We appreciate the wonderful strides that have been made in the development and manufacture of the pneumatic tire, and would be remiss indeed were we not to compliment the tire manufacturers for the good work they have done. We trust that, through research and development work, a still better tire will be offered in the future.

Suggestions Made for Changes in Practice

—T. C. Smith

American Telephone & Telegraph Co.

MR. HALE upholds the classic theme of the manufacturers that overloading tires is uneconomical. I wonder if the answer is quite as simple as this. For instance, it seems very questionable whether a load which for only 10 per cent of the time exceeds the manufacturers' table by 40 per cent would warrant buying larger tires. His exposition of this matter is very general. Also, from impact tests, it is evident that speed has a direct and very marked effect upon shock load transmitted through the tires. For this reason, why would it not be rational to give a higher rating to tires for use on comparatively slow-moving governed-trucks? This effect is in the same direction as the heat factor caused by speed, which Mr. Hale discusses.

From what study I have made of the matter, it seems that, for safety reasons, retreading should be limited to six-ply and heavier tires. Fortunately also, the greater economy from retreading is possible with the heavier tires.

Test to Determine Probable Tire Life

—F. L. Faulkner

Armour & Co.

IT has been the policy of our company for quite some time, in connection with the purchase of new equipment, to specify adequate tire size to provide the truck with sufficient capacity to handle the maximum gross of the vehicle.

With advancing prices in tires in the last two or three years, the question arose as to the advisability of oversizing and at what point we should stop in applying oversize, using the Tire and Rim Association rating in computing the maximum allowable gross weight to be carried.

In discussing this subject with both tire manufacturers and operators, the answer to this problem was not available; so, we set about to determine first-hand a measure on the probable tire life of tires carrying an overload versus a normal load, using the Rubber Association rating as a base.

We selected two trucks of the same make, model and capacity, and equipped one with 9.00-20 dual rear-tires and the other with 9.75-20 dual rear-tires from regular tire production, but cross-checked as to specifications. The tires were inflated to 5 lb. per sq. in. above the tire manufacturer's rating, and, due to high-crown roads on both routes, the outside dual tires were given an additional 5 lb. per sq. in. air-pressure. The trucks were alternated on the routes to equalize conditions, and the tires were changed from one truck to the other every thirty days. Tread-depth measurements were made at 1000 mile intervals at four places across the tread of the tire. These measurements were made in conjunction with the tire manufacturer, who set up the necessary facilities for this determination.

The test covered a period of four months. The truck in

question weighed 7356 lb. (our truck No. 11158), covered 22,568 miles and hauled 1,770,899 lb. in 122 trips, or an average of 14,560 lb. per trip. Truck No. 11308 covered 20,666 miles and hauled 1,445,255 lb. in 116 trips, averaging 12,460 lb. per trip. The average gross vehicle weight for truck No. 11158 was 21,866 lb.; the average gross vehicle weight for truck No. 11308 was 19,810 lb. The truck running with 9.00-20 tires had a gross vehicle weight of 19,700 lb., and while operating with 9.75-20 tires had a gross vehicle weight of 23,400 lb.

The trucks in question had as near ideal load-distribution as could be obtained for any conventional-type truck. However, it was not possible to take advantage of the allowable front-tire capacity. Therefore, all wear measurements were limited to the rear wheels. At the end of test the 9.00-20 tires were too smooth for further operation; the 9.75-20 tires had approximately 20 per cent more life remaining. The trucks used 3 per cent less gasoline running on the 9.75-20 tires than when operating on 9.00-20 tires due, in my opinion, to maintaining a better profile under load; also, a slight advantage in rolling radius was noted with the larger tires. The 9.75-20 tires cost approximately 25 per cent more than the 9.00-20 tires. After the test, tread wear gaged as follows:

		A	*B*	*C*	*D*
9.00-20 Tires	L R I.....	0.28	0	0	0.095
	L R O.....	0.16	0	0	0.192
	R R I.....	0.05	0	0	0.09
	R R O.....	0.095	0	0	0.130
9.75-20 Tires	L R I.....	0.33	.07	0	0.26
	L R O.....	0.275	0	0	0.35
	R R I.....	0.295	0	0	0.255
	R R O.....	0.320	0	0	0.316

We concluded that the test was sufficiently representative to indicate what we could expect in our average operation, and as a result continue to oversize above the truck manufacturers' recommendations, and assume that the Tire and Rim Association recommendations are conservative and conducive toward low operating costs per tire mile.

Explanation of Why Load Ratings Are Inconsistent

—**Burgess Darrow**
Goodyear Tire and Rubber Co.

MR. HALE'S paper is exceedingly well prepared and is no doubt extremely interesting to those who are not primarily engaged in the tire business. To those of us who are in the tire business it has also done some good to listen to a talk on some of the fundamentals of tire design.

Mr. Hale touched on six phases of tire engineering:

(1) Underlying principles of tire design with special reference to overloading.

(2) An analysis of the existing load ratings for balloon truck and bus tires.

(3) A plea for the use of balloon truck and bus tires and the consequent dropping out of the older type of so-called high-pressure tires.

(4) A suggestion for a line of tires larger and softer than the present balloon truck and bus tire and more like a certain line of Michelin tires, but Mr. Hale has carried their idea further.

(5) A discussion of various hauling conditions, requiring tires especially designed for the service.

(6) Instructions for the care of tires.

The talk on tire construction and overloading is a fine piece of work. I can endorse his plea for the use of balloon tires on trucks and the dropping of the old high-pressure type. It was well to mention new types of tires that are fast coming on the market for special classes of hauling. His remarks on tire care and upkeep were to the point.

I shall comment on existing load ratings and tire dimen-sions and add comments along Michelin lines.

On the matter of load ratings, Mr. Hale's charts may have convinced his audience that there were some terrible inconsistencies in the dimensions or loads of two or three of the tires in the series. There are inconsistencies, but they are in my opinion of very slight importance. For example, Fig. 17, in which thirteen actual-tire cross-sections were plotted against their rated loads, failed to give a smooth curve, but the curve would have been beautifully smooth if the tire industry should decide to make three of the sizes just a shade larger, increasing an 8.25 tire by ¼ in., the 10.50 tire likewise by ¼ in., and the 11.25 tire by about ⅜ in. Or, the curve could be smoothed out by decreasing the dimensions of a few other sizes; but decreases are never made.

That the load ratings are slightly inconsistent is due to a number of reasons, some of which are as follows:

(1) Each tire company has to make sure that its tire is as big as the other fellow's, and the result is a slow growth in tire sizes that does not affect all sizes alike.

(2) There have been cases where there were reasons for increasing loads in the case of one tire without disturbing the series.

(3) The tire companies for years differed on ideas of rim width. For example, whether a 7.50 tire performed better on a 6-in. rim 4.33 wide, or on a 7-in. rim 5 in. wide; likewise, should a 10.50 tire be recommended on an 8-in. rim 6 in. wide, or a 9-10 rim 7.33 wide. This is fairly well straightened out now, but it leaves its mark on the charts.

I submit that, on the whole, the Tire and Rim Association has done at least a satisfactory job on load ratings and tire sizes, considering all the factors it has had to contend with.

Second, I wish to discuss the line of extra-large soft-tires which Mr. Hale suggests in Fig. 21 and Table 2.

Taking figures from Table 2, he proposes, in place of a 9.75-20 tire, of which all have a fairly good idea, to use a 12.00-18. The proposed tire has a 2 in. smaller rim; but, in spite of this, the tire would stand 1½ in. higher in overall diameter. The proposed tire would be about 2 in. wider, which means about 11 in. more space required for tires on the axle in the case of duals, 2 in. more for each of four tires, and 1½ in. more spacing of the two rims on a wheel. For all this, the proposed tire would operate at a pressure of 50 lb. per sq. in. instead of 70 lb. per sq. in. with the present tire, and no doubt give a perceptibly better ride.

The Michelin Co. of France has had on the market for some time a line of so-called "Comfort" tires which are a little bigger than American tires. For a load rating about the same as our 9.75-20, they have a tire about ¾ in. wider; and in place of the 70 lb. per sq. in. pressure in our tires, they recommend 57 lb. per sq. in. In other words, they are much more conservative than Mr. Hale.

History is in the habit of repeating itself and, on three occasions when the tire companies have brought out new lines (I refer to balloon passenger-car tires, balloon truck-tires, and the extra-low-pressure passenger-car tires of today), they have overstepped the mark and very soon after the introduction of such tires had had to back up: first, on the amount of oversize that is practical; second, that in all preceding cases pressures have not gone as low as was first expected.

I certainly am not convinced that we have the ultimate yet in tire sizes. I do feel that the next step in bus tires, if it comes, is going to be a much smaller step than Mr. Hale is suggesting, and I base this feeling on the three previous experiences we have had.

Again I want to state that Mr. Hale's paper is a very good one. I differ from him on one or two matters; but please understand that we do not differ in theory but only in the ways of working things out, or a matter of degree.

Positions Open to 7 S.A.E. Members

The positions outlined below represent a selection of seven excellent openings among the recent listings with the Society's Placement Service. If you're interested, act now!—write the S.A.E. Placement Service, mentioning number of opportunity.

P 556: *An experienced Executive Engineer for a large airplane group, building both commercial and military ships in the East. He will be responsible for organizing and following the work of the Engineering Department and will be the contact between it and the manufacturing personnel and the consumer. He must have had similar experience and a good record. State age, education, experience; photograph desirable. All applications will be held confidential. Personal interviews will be arranged with applicants who warrant further consideration.*

E 415: *Service (Mechanical) Engineer with good technical education and practical sales viewpoint; anti-friction bearing experience preferred. Must have initiative and tact. Give age, salary desired, and complete details.*

E 485: *Automotive Engineer, with particular emphasis on automotive clutches, desired for development work. Only engineers with wide clutch experience will be considered. State experience, education, references, salary expected.*

P 594: *Engineer, 30-35 years of age, to supervise application of anti-friction bearings, both ball and roller, in automotive design; one who has had intimate association with the more progressive automobile factories in the automotive area. After six months' training at headquarters, will be located permanently in Detroit as resident engineer. Should be progressive in engineering ideas and keep abreast of the rapid development and changes taking place in automotive design.*

E 49: *Method and Equipment Manager, with thorough knowledge of plant layout, presses, jigs and fixtures, tools and dies in the automotive industry. Age between 35 and 50. Give all references in first letter.*

P 512: *Industrial Sales Engineer, about 30 years old, well grounded in mechanical engineering principles but with several years' sales engineering experience, or some sales ability. The work will be 75% engineering but a sales approach is essential. Permanent position. Moderate starting salary; traveling expenses paid.*

E 52: *Engineer, qualified to do estimating work on a miscellaneous run of tools, parts and special machinery such as a contract job shop has to estimate in the regular routine of its business for sales and quotation purposes. Should have good ground work of shop and engineering department experience, and willing to start at moderate salary. Permanent position for the right man.*

▲

The S.A.E. Placement Service operates every working day of the year as a free contact between employers and S.A.E. members who seek new connections. If you are an employer or unemployed, write the Placement Service for information which will be of interest and value to *you*.

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THE PLACEMENT SERVICE, Society of Automotive Engineers, Inc.
29 West 39th Street, New York, N. Y.

More Views on Causes of Deposits in White Metal Carburetors

RESULTS of studies by E. L. Baldeschwieler, G. M. Maverick and J. E. Neudeck, of the Standard Oil Development Co., regarding the causes of stoppage of fuel flow by powdery material in white metal carburetors were published in the October, 1935, issue of the S.A.E. Journal.

Subsequent to publication of that article, which was sponsored by the Fuels Research Subcommittee, comments and discussions of the points which it raised have come from a variety of sources.

Here are presented the views of other investigators who have sent us discussion on this subject.

Laboratory Study Made of Carburetor Corrosion

—George Calingaert
—Graham Edgar
Ethyl Gasoline Corp.

THE Laboratories of the Ethyl Gasoline Corp. have studied intermittently over a period of years the corrosion of light-metal alloys used in fuel systems of automobiles and aircraft engines. This study has covered the most widely used alloys and particularly those of magnesium, of aluminum and also of zinc-base metal used in die-casting. The results obtained confirm those of other laboratories to the effect that no corrosion takes place in the absence of water.

It is well known that chromate treatment of some of these metals will protect them against corrosion for a time which depends on the conditions of exposure. Experiments carried out in this laboratory have shown that the presence of a soluble chromate in those places where water might tend to accumulate will inhibit corrosion completely. The commonly used chromates are, however, very soluble in water and will tend, therefore, to be used up rapidly. In order to increase the life of the inhibitor, a search was made for chromates which would be just soluble enough to give satisfactory protection. Of those tested, calcium chromate, $CaCrO_4$, with a solubility of approximately 2 per cent, was found to be very satisfactory.

In order to test the efficiency of the principle, two die-cast metal fuel pumps, in which the settling bowl is integral with the casting, were selected. In this type of equipment the water and sediment which settle out of the fuel remain in contact with the metal. A small amount of water was put in each pump and both pumps were then operated intermittently to circulate gasoline for a period of five weeks. One pump was

Fig. 1—(Calingaert-Edgar Discussion) Sediment Bowl of a Die-Cast Fuel-Pump
Note the corrosion products in the lower right-hand corner.

operated as a blank, without the inhibitor, and in the second pump, two small pellets about ¼ in. in diameter of calcium chromate fluxed with borax (two parts of borax to one part of chromate) were introduced as inhibitors. Figs. 1 and 2 illustrate the condition of the pumps at the end of the test. The effect of the inhibitor is particularly noticeable in the lower right-hand corner of the sediment chamber. When the inhibitor was used (Fig. 2), the settling bowl is untarnished and particularly the fuel-outlet line which remains visible in the bottom of the settling bowl. When no inhibitor was used (Fig. 1), this same corner where the water accumulates is almost filled with the typical grayish-white corrosion product.

This method of preventing corrosion of light-metal equipment appears entirely practicable for fuel pumps, carburetors, aluminum and magnesium-alloy gasoline-tanks, etc. Various methods of utilizing the calcium-chromate inhibitor suggest themselves, such as the pellets already described, perforated containers, small bags, and the like, depending upon the design and dimensions of the equipment. The life of the inhibitor obviously depends upon the quantity used and the amount of water present, but it might easily last for the life of the equipment.

Conclusions.—Die-cast (zinc-base) metal such as is used in carburetors, fuel pumps, etc., will corrode where it remains in contact with water. This corrosion can be inhibited completely by introducing at the point at which water collects a small pellet of calcium chromate, fluxed with borax.

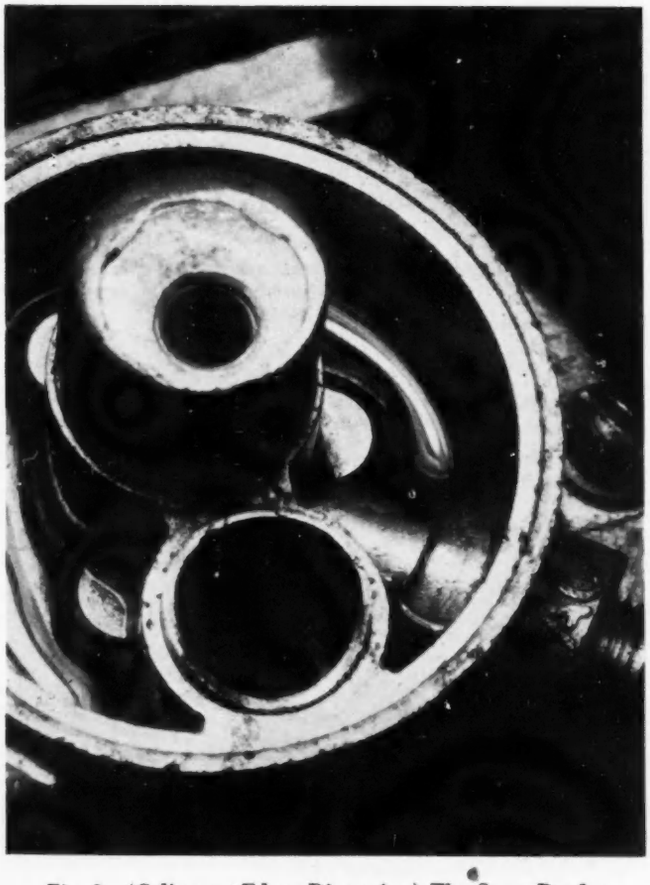


Fig. 2—(Calingaert-Edgar Discussion) The Same Bowl as in Fig. 1 When Calcium-Chromate Inhibitor Is Used
Note the clear appearance of the water and the untarnished metal.

Corrosion Almost If Not Entirely Caused by Water

—H. C. Dickinson
National Bureau of Standards

THE occurrence of white deposits in aluminum and die-cast carburetors was noted during or shortly after the World War in connection with aircraft carburetors. A somewhat crude analysis of the material made at that time indicated clearly that the products came from corrosion resulting from the presence of water.

It seems quite obvious that most of the aluminum alloys which are used in die casting are readily attacked by water and that water is commonly present in carburetor float bowls. Consequently, the difficulty is one which might almost be predicted in advance whenever materials of this kind are used.

Wide experience as well as specific tests have shown for a long time that all these materials are substantially immune from attack by gasoline, and it is satisfying to learn that the addition of tetraethyl lead to gasoline does not alter this fact.

Protective Films as a Corrosion Preventive

—C. R. Maxon
The New Jersey Zinc Co.

WE have read with keen interest a preliminary draft of this paper. The problem of fuel-line blockage has, of course, been under consideration for quite a few years. While the writers have focused their attention on the corrosion of the zinc carburetors, it should not be overlooked that the other factors such as the presence of lint and other foreign materials and of corrosion products from iron, etc., enter into the picture to a very considerable extent. The manufacturers of carburetors and fuel pumps have taken into consideration the possibility that the white corrosion products produced on zinc by water present in gasoline may, under some circumstances, cause some trouble in the fuel system and have as a precautionary measure adopted a number of protective films as a means of preventing the development of these white salts. The majority of the carburetors installed on 1935-model cars have one or another of these films applied to them. It is probably safe to say that today any possibility of fuel-line stoppage resulting from the use of zinc carburetors has been eliminated.

Uses Zinc Dicromate Coating on Die Castings

—E. O. Wirth
Bendix Products Corp.

WE have been using a zinc dicromate coating on all our zinc die castings. This material retards corrosion and to date is the best preventative.

What Members Are Doing

Miller McClintock, director, Bureau for Street Traffic Research, Harvard University, addressed the annual meeting of the American Standards Association, Dec. 11, on the subject of highway safety. F. E. Moskovics, vice-president of the A.S.A., presided.

William S. Knudsen, executive vice-president, General Motors Corp., was the principal speaker at the Fourth Annual Slide Rule Banquet Nov. 19 at the Aztec Tower of the Guardian Building sponsored by the seven student engineering societies at the University of Detroit. S.A.E. President William B. Stout was toastmaster.

Jerome A. Fried is now owner of the Ithaca Scientific Instrument Co. of Ithaca, N. Y. He was formerly chief engineer of the Allen Walls Adding Machine Corp. of the same city.

Warren F. Teigeler, formerly technical clerk with W. R. Grace & Co., has joined the Fafnir Bearing Co., New Britain, Conn.

Del S. Harder, formerly on the staff of the vice-president in charge of operations of Fisher Body Corp., has been transferred to Grand Rapids, Mich., as resident manager of a new plant to be constructed for the Grand Rapids stamping division of General Motors Corp.

H. Wirshing, Western representative, Waukesha Motor Co., has been transferred from San Francisco to Los Angeles.

W. C. Stevens, who has been chief engineer of Cutler-Hammer, Inc., recently was elected vice-president in charge of the engineering department of the same company.

M. C. Van Sandwyk has joined General Motors South Africa, Ltd., as head of their plant service school located at Port Elizabeth, So. Africa. He was formerly a partner in the firm of Van Sandwyk, Edwards (Pty.) Ltd.

Captain Edward V. Rickenbacker, vice-president and general manager, Eastern Air Lines, has been in Europe conferring with officials of the principal European airways on details of the international cooperation which is expected to make possible a regular transatlantic air service within the next three years.

Loyal George Tinkler, formerly superintendent of the metallurgical department, S. S. White Dental Manufacturing Co., has joined the Vanadium Corp. of America as sales promotion metallurgist.

William B. Ross, who was previously head of the chemical engineering section of the Socony-Vacuum Oil Co., Paulsboro, N. J., is now with the Pure Oil Co., Chicago, Ill.

James Lynah, director of athletics, Cornell University and a member of the Society since 1922, will address the Cornell Club of Michigan at a luncheon to be held in Detroit, Jan. 11.

Sherrod E. Skinner, who for the last two years has been assistant general manager of the Ternstedt manufacturing division of General Motors, was recently appointed general manager. Mr. Skinner came to Ternstedt as chief

S. E. Skinner

engineer in 1930. For ten years prior to that he was employed by Landers, Frary and Clark, manufacturers of household appliances.

Ralph E. Flanders, president, Jones and Lamson Machine Co., Springfield, Vt., has been elected a director of the National Life Insurance Co., Montpelier, Vt.

Henry M. Crane, technical advisor to the president, General Motors Corp., was recently at the wheel of a Crane Simplex, which he designed and built, as it turned its 300,000th mile. The car was built for H. L. Satterlee, New York lawyer, in 1915 and it has been in use virtually every day of its twenty years of life.

C. Lincoln Christensen, formerly experimental engineer with the Anglada Motor Corp., is now technical editor of *Automobile Trade Journal,* Philadelphia, Pa.

Walter C. Trautmann is now layout draftsman with the Kinner Airplane and Motor Corp., Glendale, Calif.

W. H. Beal, formerly vice-president, Cord Corp., is now board chairman of the New York Shipbuilding Corp., Camden, N. J.

W. H. Beal

C. H. Cuno, president, Cuno Engineering Corp., and *H. L. Sharlock,* director of public relations, Bendix Aviation Corp., are serving as members of the Motor and Equipment Manufacturers Association board of directors for 1936.

N. R. Chandler, who has been managing director, Allied Motors, Ltd., Bombay, India, is now affiliated with Rolls-Royce, Ltd., Derby, England.

Earle G. Fahrney is in the engineering tests department of the General Motors Proving Ground, Milford, Mich.

Taro Mori, formerly airplane engine designer, Tokyo Gas & Electrical Engineering Co., Ltd., has joined the Nippon Diesel Industry Co., Ltd., Tokyo, Japan, as assistant chief engineer.

W. A. Maynard, formerly factory representative, Cummins Engine Co., Columbus, Ind.,

Walter A. Maynard

is now Cleveland branch manager of the White Motor Co.

E. N. Klemgard, with the Shell Oil Co., Martinez, Calif., refinery, has recently been appointed technologist. He was formerly manager of the motor laboratory.

Foster M. Gruber, previously with the Wright Aeronautical Corp., Paterson, N. J., as test engineer, has joined the Kinner Airplane and Motor Corp., Ltd., Glendale, Calif., as engineer.

Walter M. Lipps has joined the Keasbey & Mattison Co., Ambler, Pa., as automotive engineer doing both sales and engineering work

Walter M. Lipps

pertaining to standard equipment. He was previously automotive engineer for the Russell Manufacturing Co.

Lieut.-Col. James R. Hill has moved his headquarters from Manila, P. I., to Fort William McKinley, Rizal, P. I.

Edward P. Warner

Edward P. Warner has recently opened offices as a consulting engineer in New York. He has resigned as editor of *Aviation* which position he held since 1930. He was, for three years prior to that, Assistant Secretary of the Navy for Aeronautics. He was vice-chairman of the Federal Aviation Commission which reported to the President in 1935.

Mr. Warner is a past-president of the Society, having held that office in 1930. He has been prominent in the activities of the S.A.E. since 1923. At the present time he is a member of the Aircraft Activity Committee, the Research Committee, the Riding Comfort Subcommittee, the Wright Brothers Medal Award Committee and is the S.A.E. representative on the Engineering and Industrial Division of the National Research Council.

W. R. Herfurth, formerly assistant secretary-treasurer, New York State Code Authority for the Trucking Industry, has joined the Seaboard Freight Lines, Inc., as operations and maintenance manager.

Franz Oberle, sales engineer, Diesel division, United American Bosch Corp., has been transferred to Detroit from Chicago.

Fred W. Cederleaf

Fred W. Cederleaf, who was previously standards manager, General Motors of Canada, Ltd., is now affiliated with the Ex-Cell-O Aircraft and Tool Corp., Detroit, Mich., as manager, machine tool division.

Ludger E. La Brie has joined the Chrysler Corp. at Highland Park, Mich., recently as designing engineer. He was previously development engineer with the Bendix Brake Co.

Arthur D. Wolfe, who has been vice-president and general manager of Motor Valve Products Corp., Ravenna, Ohio, has recently joined the Bellevue Manufacturing Co., Bellevue, Ohio, as secretary.

Annual Meeting Exhibitors

AN engineering exhibit for engineers, in charge of engineers, will be held Jan. 13 to 17 in the Book-Cadillac Hotel, Detroit, as one of the features of the S.A.E. Annual Meeting. A partial list of exhibitors appears below, in alphabetical order. Visit the exhibits when you attend the Annual Meeting.

Acheson Colloids Corp.
Aetna Ball Bearing Manufacturing Co.
Aluminum Co. of America
American Cable Co., Inc.
American Chemical Paint Co.
Bantam Ball Bearing Co.
Black & Decker Mfg. Co.
Campbell, Wyant and Cannon Foundry Co.
Carborundum Co.
Cleveland Graphite Bronze Co.
Cities Service Co.
Continental - Diamond Fibre Co.
The Deluxe Products Corp.
Doehler Die Casting Co.
The Dole Valve Co.
H. A. Douglas Manufacturing Co.
Harrison Radiator Corp.
Hercules Motors Corp.
Inland Manufacturing Co.
International Nickel Co., Inc.
Micromatic Hone Corp.
Monroe Auto Equipment Co.
Oakite Products, Inc.
Tinius Olsen Testing Machine Co.
Parker-Kalon Corp.
Spicer Manufacturing Corp.
Spring Washer Institute
Timken Roller Bearing Co.
United American Bosch Corp.
Van Dorn Electric Tool Co.
Victor Manufacturing & Gasket Co.
John Warren Watson Co.
Waukesha Motor Co.
The Yale and Towne Manufacturing Co.

Erratum

In the report of the Pacific Coast Meeting published in the December, 1935, S.A.E. JOURNAL in Column 2, page 33, the following statement is made in lines 10 and 11: "The company specifications call for fuels not leaded more than 3 cc. per gal."

The amount was inadvertently printed as "3 cc. per gal.," whereas it should have been printed: "not leaded more than 2 cc. per gal."

George Spatta, until recently chief engineer, has been made vice-president and general manager of the Clark Equipment Co.

Raymond A. Bittner has joined the Ralb Aircraft Co., Chicago, as engineer. He was, until recently, engineer with the Smith Aircraft Co.

John J. Nargi is now employed in the engineering department of the International Motor Co., Plainfield, N. J. He was previously with the Continental Motors Corp. as draftsman.

Robert Motion is now marketing assistant with the West India Oil Co., S.A., Port of Spain, Trinidad, B.W.I. He was formerly assistant manager of the same company.

Walter F. Roeming is with International Harvester Co. as engine layout man. He was previously draftsman with the Buda Co.

Hans A. Wagner, formerly a member of the Student Branch at New York University, is office assistant with the U. S. Fidelity and Guaranty Co.

Carl Behn is affiliated with Otto Engine Works as Philadelphia sales manager. He was previously sales engineer with the United American Bosch Corp.

Charles B. Jahnke has recently affiliated with the Cooper-Bessemer Corp., Mt. Vernon, Ohio, as executive engineer. He was previously research engineer with International Harvester Co., Chicago.

Walter O. Briggs, president, Briggs Manufacturing Co., having exercised his option to buy the half interest of the late Frank S. Navin in the Detroit Tigers Ball Club, is now the sole owner.

The National Association of Motor Bus Operators has recently elected the following S.A.E. members as officers: *E. D. Merrill* will be treasurer for 1935-1936, *J. F. Winchester, George M. Kryder,* and *C. W. Stokes* were elected associate member directors; and *H. F. Fritch* a member of the board of directors.

Edford M. Walter

Edford M. Walter, assistant sales manager, Ethyl Gasoline Corp., was killed in an airplane accident near Nunda, N. Y., on Dec. 11. His pilot, Dewey L. Noyes, was also killed when their airplane crashed in a fog.

Mr. Walter was 41 years of age. He had been a member of the Society since 1923, at which time he was affiliated with the General Motors Research Corp. In 1925 he joined the Ethyl Gasoline Corp.

He was graduated from the Kentucky State University with the degree of Bachelor of Mechanical Engineering.

Carl Herskind

Carl Herskind, a member of the Society since 1928, died on Nov. 1. Mr. Herskind was active in automotive service work for a number of years. Prior to opening his own garage in Oakland, Calif., in 1929 he was, for nine years, superintendent of the Pacific Nash Motor Co. in Berkeley, Calif.

Surviving is his wife, Mrs. Lillie Herskind

Papers from Recent Meetings in Digest

HERE are digests of papers presented at recent Regional and Section Meetings of the Society.

* * * *

Some of these papers will be printed in full in the S.A.E. JOURNAL.

Mimeographed copies of *all of them* will be available, until current supplies are exhausted, at a cost of 25 cents per copy to members; and at 50 cents per copy to non-members, plus 2% sales tax on those delivered in New York City. Orders for mimeographed copies must be accompanied by remittance and should be addressed to Sessions Secretary, Society of Automotive Engineers, 29 West 39th St., New York.

Pacific Regional Meeting Papers—San Francisco, Nov. 18-19

Some Practical Gasoline and Oil Problems in Transpacific Flying—*John C. Leslie, acting division engineer, Pan-American Airways System, Pacific Division.*

SOME of the practical fuel and lubricating problems of the Pan American Airways System transpacific project are outlined briefly by the author.

The major problem is gasoline economy, that is, burning fewer pounds of gasoline per horsepower hour. The immediate problem is to secure engines for the service that can operate indefinitely under cruising conditions with specific fuel consumptions between 0.40 and 0.42 lb. per hp-hr. A safe lowering of specific consumption from 0.48 to 0.42 lb. per hp-hr. means a 12.5 per cent increase in payload.

The most fundamental problem at present in connection with lubricating oil is how to measure the actual lubricating value of a given oil.

Other features include comments on engines of higher output and sludge formation. Mention is made of the miscellaneous oils and lubricants used on an airplane like the China Clipper.

Automotive Fleet Maintenance Problems on the Pacific Coast—*S. B. Shaw, automotive engineer, Pacific Gas and Electric Co.*

AFTER describing the geographical, physical and climatic conditions under which fleet motor-vehicles must operate, the author states that the manner in which maintenance is provided for can generally be divided into two classes, fleet-shop maintenance and outside-shop maintenance. Those fleets with high density as measured by vehicles per square mile of territory as a rule operate their own shops, while those with lower densities usually depend upon the outside shops, operating under contract systems.

Usually, maintenance work is decentralized, distances between shops varying from 20 to 500 miles, again depending on density. One fleet of nearly 2000 vehicles in an area approximating 100,000 square miles (relatively high density) operates 15 shops where overhauling is done, besides an equal number where minor repairs and servicing are taken care of.

Several fleets employ traveling mechanics or mechanic inspectors. These men travel by automobile or truck, taking with them such tools and supplies as are required and inspect and/or maintain a group of vehicles within a radius of 50 to 100 miles. This system is commonly used where density or concentration does not warrant installation of shops. These men obtain necessary parts locally. In some cases they supervise and direct work done by local outside shops or advise when vehicles are in need of transfer to base shops for more extensive work.

With this background, the more detailed and specific maintenance problems that are constantly arising and which devolve upon the fleet maintenance man for solution are considered. They are fuel, lubrication, engine, general chassis, body and finish, and tire problems.

Diesel Fuel-Knock Testing—*T. B. Rendel, charge automotive research laboratory, Shell Petroleum Corp.*

IN the last few years the very marked tendency toward increasing the speed of Diesel engines has enabled this type to compete with the gasoline engine in road-transport work. This has brought to the fore a phenomenon known as "Diesel knock" and the fact that these newer types of engine are somewhat more sensitive to fuel changes than their more elderly cousins.

After reviewing the history of the Cooperative Fuel Research compression-ignition engine the author outlines the method of conducting the fuel-knock tests, stating that it is possible to use both the critical-compression-ratio method of rating fuels described by Messrs. Pope and Murdock and also, in conjunction with a modified type of bouncing-pin, the delay-period method suggested by Messrs. Boerlage and Broeze, and to obtain measurements of the ignition quality of a Diesel fuel with reasonable accuracy and with a minimum amount of engine maintenance.

Recently, a series of cooperative-test samples were circulated among the members of the Volunteer Group for C.I. Fuel Research, and the check tests on this series of samples is surprisingly good considering the short length of time which the members have been operating these engines.

It is the consensus of general opinion that the results should be expressed in terms of standard reference-fuel, and the fuels most widely used as reference fuels are cetane and alpha methylnaphthalene.

Gasoline Knock Testing—*C. F. Becker, supervisor, motor laboratory, Associated Oil Co.*

WONDERFUL advancement has been made in the art of gasoline antiknock testing over a period of years; however, groups of correlation samples submitted to various laboratories for octane number often show a variation in agreement and the magnitude of deviation seems to vary according to the type of fuel supplied and octane range involved.

During the last few years octane rating has become an important

test for evaluating the quality of gasoline. Therefore, from a manufacturing standpoint, it is highly desirable to maintain precise octane control of refinery stocks.

With the present procedure and equipment for determining octane numbers, there are periods when consistent operation is difficult and results unreliable. Under these conditions laboratories are often at a loss as to what should be done to bring their equipment back into line. For this reason operators of C.F.R. equipment on the West Coast were contacted and a survey of their operating experiences obtained. The material collected in this survey is the basis for this paper. The items mentioned are to be considered merely a discussion of C.F.R. engine testing of West Coast laboratories.

From the West Coast laboratory survey it was the consensus of opinion that, basically, the C.F.R. engine is a satisfactory apparatus for determining octane numbers; however, some laboratories expressed the opinion that more accurate and consistent results may be obtained by (*a*) continued refinement of engine and instrumentation; (*b*) establishment of a more precise test procedure; and (*c*) universal adoption of standardized reference fuels.

The C.F.R. engine is discussed and the instrumentation used in the West Coast laboratories for testing is described. The laboratory procedure is stated.

Regarding fuels, the West Coast survey showed that a large number of reference fuels were found to be in use, and this undoubtedly is one of the causes of varying results experienced in correlation work.

Extreme Pressure Lubricants Testing—*G. L. Neely, research engineer, Standard Oil Company of California.*

THIS paper describes briefly why extreme-pressure lubricants are necessary and what machines have most commonly been used to evaluate them.

Examples of differences in test results caused by differences in operating conditions, by the type of machine used, and by the method of installation of the machine, are presented.

The new S.A.E. Extreme-Pressure-Lubricants Tester is described, and the probable advantages of the machine are discussed.

Other requirements of commercial extreme-pressure lubricants are listed.

In conclusion, the statement is made that the new machine is believed to constitute an approach to the gear problem that is more sound fundamentally than that of any of the other machines that have been used. In the older machines one or more of the test surfaces was always stationary while, in the S.A.E. Tester, both surfaces are in motion and a combination of sliding and rolling approaching that of gearing may be achieved.

In spite of this, however, the problem is by no means solved and a true solution can only be achieved by correlating the S.A.E. Tester with actual service tests.

Cooperation—The Keystone of S.A.E. Research—*C. B. Veal, research manager, Society of Automotive Engineers, Inc.*

THE research work of the Society requires the active participation of as many members as possible. The type of research project undertaken is adapted to cooperative work.

David Beecroft, president of the Society at the inauguration of its research activity, distinguished between three types of research: explorational or purely scientific; intensive research in the fields brought within the ken of engineering by explorational research; and industrial or development research.

The S.A.E. Research Committee and its subcommittees embrace a membership of 150 engineers from the automotive and petroleum industries.

The Society's research procedure is now in its third stage of development—that of cooperative experimental investigation. The first stage was accumulation and dissemination of technical information and the second stage involved the formulation of programs and the designation of impartial research laboratories to carry them out.

Advantages of S.A.E. research procedure and several typical research projects were shown. A resumé of current projects concluded the paper.

Standards Road to Profits—*R. S. Burnett, standards manager, Society of Automotive Engineers, Inc.*

THE author summarizes his paper by stating that Standardization, starting with comparatively simple applications to obtain interchangeability of parts in a few important items, has steadily grown in volume, scope and application into the present broad and comprehensive system. From a function of elemental engineering, it has expanded into a factor affecting the primary supplier, the engineer, the manufacturer, the seller, the buyer, the financer, the user and the servicer, in fact practically all who handle or use automotive products. It is a factor in labor relations, legislation and regulation, and is woven into practically all the elements of modern life.

This development has taken place entirely because standardization has proved its value in reduced waste, fewer varieties, conserved effort, lower costs, improved usage and simplified service.

Where it was originally used by technical groups, it is now an important factor in trade, business, government, commerce, science and practically all ordinary walks of life.

It is a generally accepted fact that mass production and efficient operation would be practically impossible except at greater costs if we did not have the benefit of these standardization activities.

Oregon Section Paper

Friday, September 13

Safety—*Fred L. Dennis, director of highway safety, Bendix Products Corp.*

THE paper is replete with entreaties and suggestions for reducing the terrible accident toll caused by motor-vehicles and their drivers, accident statistics and citations of specific motor-vehicle accidents reported in the daily press being used to emphasize the extreme necessity for motor-vehicle safety-inspection.

Inadequate penalities imposed on reckless and intoxicated drivers are deplored, methods used by the police accident-investigation men are cited, and it is stated that motor-vehicle safety-inspection, especially as applied in municipalities under ordinance, is a plan to maintain in all motor-vehicles the original safety built into them by the manufacturers.

Milwaukee Section Paper

Monday, October 7

Anti-Friction-Bearing Applications in Passenger Cars, Trucks and Buses—*E. Wooler, chief engineer, Timken Roller Bearing Co.*

CALLING attention to the fact that the automotive industry is by far the largest user of anti-friction bearings, the author states that, because the elimination of friction throughout all parts of the driving mechanism is of the utmost importance, it is necessary that the bearings used require the minimum amount of attention and that the required bearing capacity at any point of application be provided for in the least possible space. These requirements make the application of an anti-friction bearing necessary at practically every point in the chassis where a bearing is required.

Illustrations are given to the common types of anti-friction bearings and comments are made on how and where they are used, together with statements of their advantages and disadvantages.

Specific applications are then analyzed, such as front-wheel bearings, rear-wheel-axle drive, pinion mountings, and so on.

In conclusion, "mirror finish" for bearings, a distinctive grinding operation and not polishing or lapping, is described briefly.

So. California Meeting Paper

Friday, October 11

An Elementary Introduction to the Analysis of the Automobile Steering Geometry—*Frank N. Bonta, Los Angeles, Calif.*

AT present there are many systems of automobile wheel and steering-gear alignment practiced in the automotive-service field. However, it is logical to suppose that any system which does not take into account the geometric relations of the steering-gear members, with respect to the ground gear members, in the alignment treatment, in both the straight ahead and curvilinear steering, will invariably result in an unsatisfactory wheel and steering-gear alignment.

It is not the purpose of this paper to compare the relative merits of the various systems which are employed in the wheel-alignment branch of the automotive-service field; but, instead, the purpose is to present an analysis of the steering geometry in an elementary, fundamental manner, so that the underlying principles involved in the steering-alignment problem of an automobile, in both the straight ahead and curvilinear paths, may be better understood and appreciated by both the service men and the general commercial operators alike. The derivation of the working equations is also presented.

Detroit Section Papers

Monday, November 11

Fundamentals of Recent Spring-Suspension Developments—*N. E. Hendrickson, vice-president, Mather Spring Co.*

THE value of the suspension springs of an automobile lies mainly in the way that they *increase the time* in which the energy is absorbed and released, for the *gradual* dissipating of the shocks of the road relieves them of their objectionable qualities. The maker of suspension springs has an unusual problem in that he must design for large and repeated deflections, which is quite the opposite of the frame maker, who considers deflection an evil, and must design for rigidity.

The author first presents and analyzes the formula for the period of oscillation of a spring. He states that two factors are most potent in "wearing out" a spring; maximum or "top" stress, and stress range. Further mathematical analyses are then made.

In conclusion the author states that the introduction of the new types of suspension has given tremendous impetus to improvement in riding qualities, from which every motor-car user will benefit. The riding qualities of the modern automobile are so far superior to those made as recently as 1933 that the advance seems almost unbelievable. The principles of good suspension are now definitely understood by a number of engineers, and further progress should be comparatively easy.

Engineering versus Boondoggling—*C. E. Wilson, vice-president, General Motors Corp.*

ALTHOUGH the author is sure that all have a definite idea of what is meant by "Engineering," he has found many people who do not understand the term "Boondoggling" and therefore traces its origin and meaning, quoting various authorities.

The present meaning of the term "Boondoggling" covers useless ineffective activities such as making useless articles out of discarded material and doing useless things where the time and money spent are out of all proportion to the social value of the result, according to the author. He then contrasts the useless ineffective work of "boondoggling activities" with progressive forward-looking "engineering activities."

Pittsburgh Section Paper

Tuesday, December 3

Factors Affecting Cooling-System Design and Maintenance—*F. M. Young, president and general manager, Young Radiator Co.*

IT is the purpose of the engine cooling system to carry away the heat from the cylinder walls and the combustion-chamber heads. If no means of cooling were provided the high temperatures within the cylinders would soon render the engine inoperative. But not alone should it suffice to cool the engine independent of atmospheric conditions, it is necessary that, as well as possible, a uniform temperature be maintained at the cylinders. In fact, the aim should be to devise a system which would automatically maintain a fixed temperature of the cylinder walls at all times and independently of outside weather conditions; it would mean not only to cool the engine but to heat it when the atmospheric temperature conditions require it. Although many attempts have been made in this direction, the problem of maintaining a constant temperature has not yet been solved especially in regard to prevention of low temperatures. Various devices have been developed to relieve this situation. Some of these are mentioned and commented upon.

Among the subjects treated are the thermosyphon system, circulating pumps, oil coolers, temperature-control systems, radiator design, and engine design. These are treated at length, together with other important factors affecting cooling.

Meetings Calendar

S. A. E. Annual Meeting

Jan. 13-17, 1936
Book-Cadillac Hotel, Detroit

Baltimore—Jan. 9

Engineers Club; dinner 6:30 P. M. Safety meeting in cooperation with the Baltimore Safety Council. Speaker—Ambrose Ryder, vice-president, American Indemnity Co.

Buffalo—No Meeting

Canadian—Jan. 16

Royal York Hotel, Toronto; dinner 7:00 P. M. Lubrication meeting. Speakers—E. W. Upham, chief metallurgist, Chrysler Corp.

Chicago—No Meeting

Cleveland—No Meeting

Dayton—Jan. 14

National Cash Register School House; meeting 8:00 P. M. Joint meeting with the Dayton Engineers Club. The Armistice Day Stratosphere Flight—Capt. Albert W. Stevens, U. S. Army.

Detroit—Jan. 13 to 17

Book-Cadillac Hotel. Participation in Annual Meeting of the Society.

Indiana—Jan. 23

The Athenæum, Indianapolis; dinner 6:30 P.M. Production Meeting on the subject of Milling, Broaching and Grinding; speaker from the Cincinnati Machine Tool Co.

Metropolitan—Jan. 20

The Roger Smith, 40 E. 41st St., New York City; dinner 6:30 P. M. Motorboat Show Meeting.

Milwaukee—No Meeting

New England—Jan. 14

Walker Memorial, Massachusetts Institute of Technology, Cambridge, Mass.; dinner 6:30 P. M. Spring, Tire and Shock Absorber Testing Developments—S. Ward Widney, director of engineering service, B & J Auto Spring Co., Inc.

Northern California—Jan. 14

Engineers Club, San Francisco; dinner 6:30 P.M. Advance Automotive Material and Testing Equipment of Aberdeen Proving Ground—Major L. A. Miller, Ordnance Department, U.S.A.; Servicing of Bearings—Mr. Goodrich, New Departure Bearing Co.

Northwest—Jan. 17

Mayflower Hotel, Seattle, Wash.; dinner 6:30 P.M. Design and Construction of Frameless Tank Trailers for Bulk Hauling—F. V. Bistrom, Isaacson Iron Works.

Oregon—Jan. 10

Multnomah Hotel, Portland; dinner 6:30 P. M. Oregon State College Night. Effect of Gasoline Volatility on Engine Performance—Richard Wagner (student); Knock-Rating of Diesel Fuels—Terry Platt (student); Polymer Gasoline—John Comfort (student).

Philadelphia—Jan. 8

Philadelphia Auto Trades Association, Inquirer Bldg.; dinner 6:30 P. M. Engine Testing and Its Relation to Flight—Lieut. A. E. Dupuy.

Pittsburgh—Jan. 21

Hotel Webster Hall; dinner 6:30 P. M. Future Cars, including Streamlining, Weight Distribution, Engine Development—Amos Northrup, chief designer, Murray Body Corp.; Streamlining—H. Ledyard Towle, director creative design, Pittsburgh Plate Glass Co.; Automotive Beauty—Homer St. Gaudens, director Carnegie Art Galleries; Engine Design—Murray Fahnestock, automotive editor; Future Fuels and Their Influence on Future Cars—Dr. W. A. Gruss, in charge of Petroleum Research, Mellon Institute.

Southern California—Jan. 10

Mona Lisa Restaurant; dinner 6:30 P.M.

Southern New England—Jan. 16

Bond Hotel, Hartford, Conn.; dinner 7:00 P.M. Springs—Their Manufacture and Quality Requirements—J. E. Andrew, president and general manager, Wallace Barnes Co.

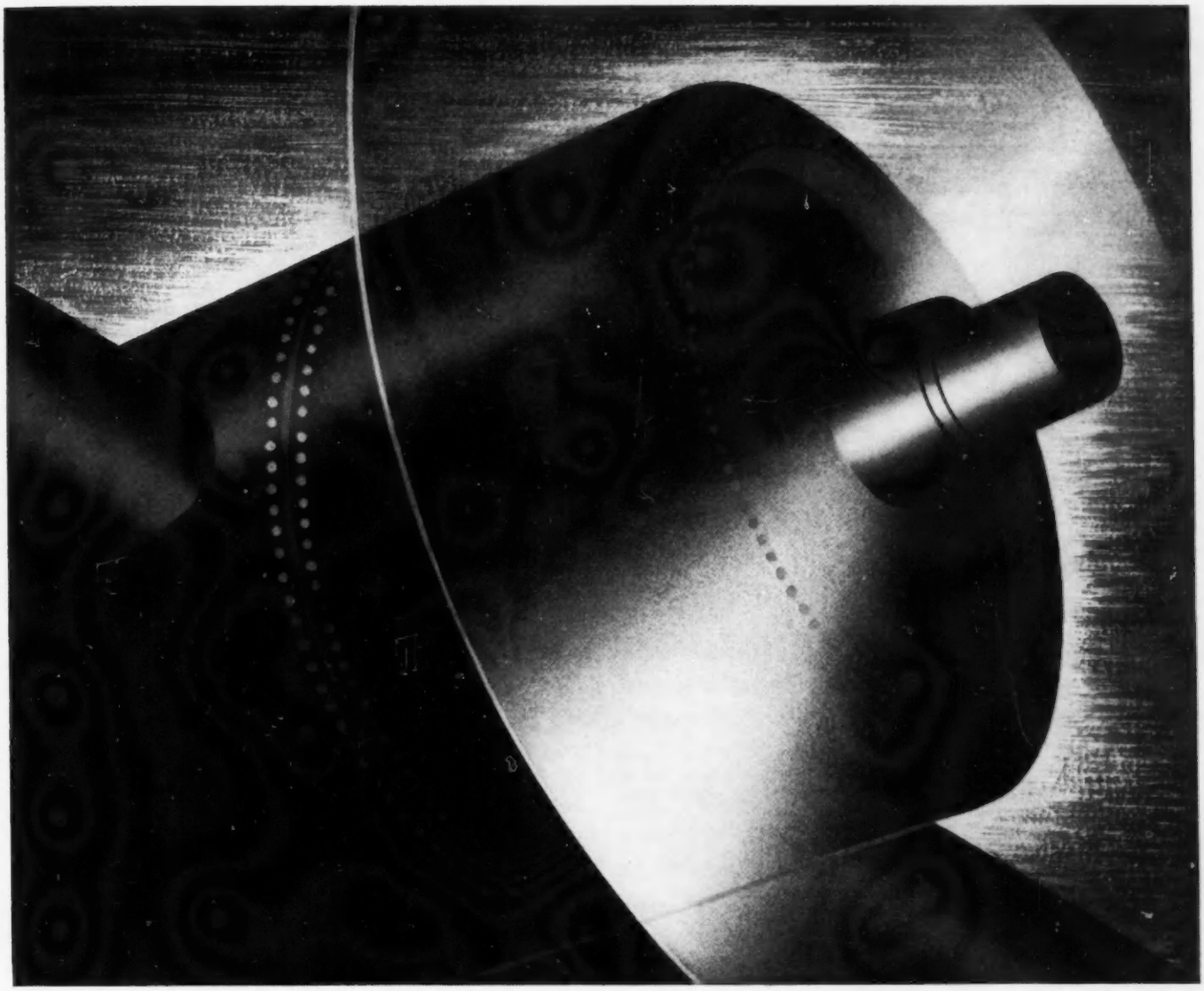
St. Louis—Jan. 10

Coronada Hotel; dinner 6:30 P. M. Diesel Fuel Pumps—by a representative of United American Bosch Corp.

Syracuse—No Meeting

Washington—Jan. 6

The University Club, 15th & I Sts., N. W., Washington, D. C.; dinner 6:30 P. M. Subject—Lubrication.



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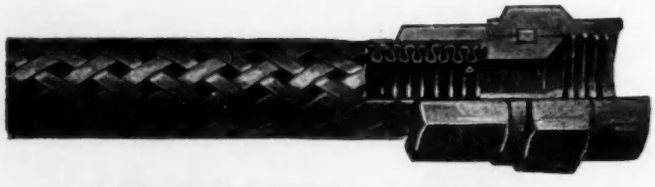

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Notes and Reviews

THESE items, which are prepared by the Research Department, give brief descriptions of technical books and articles on automotive subjects. As a rule no attempt is made to give an exhaustive review, the purpose being to indicate what of special interest to the automotive industry has been published.

The letters and numbers in brackets following the titles classify the articles into the following divisions and subdivisions: *Divisions*—A, Aircraft; B, Body; C, Chassis Parts; D, Education; E, Engines; F, Highways; G, Material; H, Miscellaneous; I, Motorboat; J, Motorcoach; K, Motor-Truck; L, Passenger Car; M, Tractor. *Subdivisions*—1, Design and Research; 2, Maintenance and Service; 3, Miscellaneous; 4, Operation; 5, Production; 6, Sales.

AIRCRAFT

Principal Effects of Axial Load on Moment-Distribution Analysis of Rigid Structures

By Benjamin Wylie James. N.A.C.A. Technical Note No. 534, July, 1935; 46 pp., 11 graphs, 20 figs. [A-1]

The Effect of Depth of Step on the Water Performance of a Flying-Boat Hull Model, N.A.C.A. Model 11-C

By Joe W. Bell. N.A.C.A. Technical Note No. 535, July, 1935; 8 pp., 34 figs. [A-1]

The 6-Foot-4-Inch Wind Tunnel at the Washington Navy Yard

By G. L. Desmond and J. A. McCrary. N.A.C.A. Technical Note No. 536, August, 1935; 13 pp., 11 figs. [A-1]

A General Tank Test of N.A.C.A. Model 11-C Flying-Boat Hull, Including the Effect of Changing the Plan Form of the Step

By John R. Dawson. N.A.C.A. Technical Note No. 538, August, 1935; 9 pp., 36 figs. [A-1]

Aerodynamic Characteristics of Several Airfoils of Low Aspect Ratio

By C. H. Zimmerman. N.A.C.A. Technical Note No. 539, August, 1935; 7 pp., 11 figs. [A-1]

A Deflection Formula for Single-Span Beams of Constant Section Subjected to Combined Axial and Transverse Loads

By Walter F. Burke. N.A.C.A. Technical Note No. 540, September, 1935; 24 pp., 24 figs. [A-1]

The Effect of the Angle of Afterbody Keel on the Water Performance of a Flying-Boat Hull Model

By John M. Allison. N.A.C.A. Technical Note No. 541, September, 1935; 10 pp., 46 figs. [A-1]

The Initial Torsional Stiffness of Shells with Interior Webs

By Paul Kuhn. N.A.C.A. Technical Note No. 542, September, 1935; 15 pp., 15 figs. [A-1]

A Preliminary Determination of Normal Accelerations on Racing Airplanes

By N. F. Scudder and H. W. Kirschbaum. N.A.C.A. Technical Note No. 537, August, 1935; 8 pp. [A-1]

The Compressibility Burble

By John Stack. N.A.C.A. Technical Note No. 543, October, 1935; 10 pp., 10 figs. [A-1]

An Application of the Von Karman-Millikan Laminar Boundary-Layer Theory with Experiment

By Albert E. von Doenhoff. N.A.C.A. Technical Note No. 544, October, 1935; 11 pp., 9 figs. [A-1]

Experiments with Suction-Type Wings

By O. Schrenk. Translated from *Luftfahrtforschung*, Vol. 12, No. 1, March 28, 1935; Verlag von R. Oldenbourg, München und Berlin. N.A.C.A. Technical Memorandum No. 773, August, 1935; 33 pp., 34 figs. [A-1]

(*Continued on page 44*)

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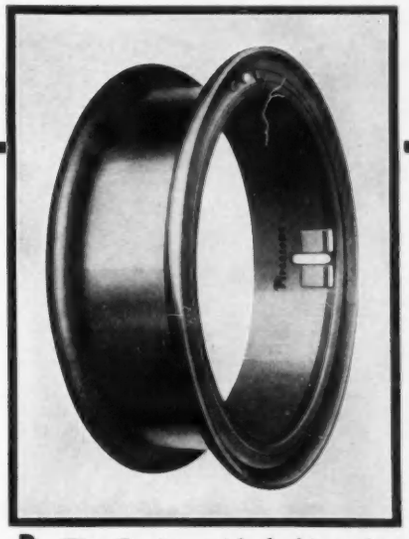

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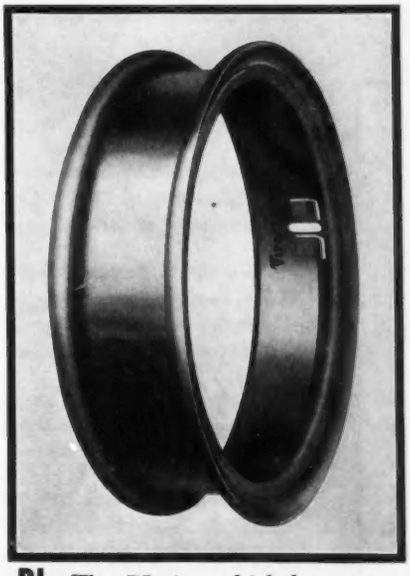

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NOTES AND REVIEWS

Continued

Tension Fields in Originally Curved, Thin Sheets During Shearing Stresses

By H. Wagner and W. Ballerstedt. Translated from *Luftfahrtforschung*, Vol. XII, No. 2, May 16, 1935; Verlag von R. Oldenbourg, München und Berlin. N.A.C.A. Technical Memorandum No. 774, August, 1935; 11 pp., 8 figs. [A-1]

Total-Head Meter with Small Sensitivity to Yaw

By G. Kiel. Translated from *Luftfahrtforschung*, Vol. XII, No. 2, May 16, 1935; Verlag von R. Oldenbourg, München und Berlin. N.A.C.A. Technical Memorandum No. 775, August, 1935; 6 pp., 15 figs. [A-1]

A Discussion of the Several Types of Two-Stroke-Cycle Engines

By Herbert J. Venediger. Translated from *Automobiltechnische Zeitschrift*, Nos. 19 and 20, October 10 and 25, 1934. N.A.C.A. Technical Memorandum No. 776, September, 1935; 28 pp., 50 figs. [A-1]

On the Wall Interference of a Circular Wind Tunnel

By Kwan-ichi Terazawa. Report No. 122 of the Aeronautical Research Institute of the Tokyo Imperial University; June, 1935; 5 pp. [A-1]

The Wall Interference of Wind Tunnels with Boundaries of Circular Arcs

By Kazuo Kondo. Report No. 126 of the Aeronautical Research Institute of the Tokyo Imperial University; August, 1935; 66 pp., with tables and charts. [A-1]

ENGINES

Hydrogen as an Auxiliary Fuel in Compression-Ignition Engines

By Harold C. Gerrish and Hampton H. Foster. N.A.C.A. Report No. 535, 1935; 16 pp., illustrated. Price, 10 cents. [E-1]

JS-Diagrams for Combustion Gases of Correct and Weak Mixtures

By Keikichi Tanaka and Seiichi Awano. Report No. 128 of the Aeronautical Research Institute of the Tokyo Imperial University; September, 1935; 15 pp., 12 diagrams. [E-1]

Combustion Characteristics in Compression-Ignition Engines

By J. Millar Logie. Published in the *Journal of the Institution of Automobile Engineers*, Aug.-Sept., 1935, p. 21. [E-1]

This paper deals with the process of combustion as it occurs in compression-ignition engines and shows how the energy released from the fuel is converted to useful power to the best advantage, indicating how the most satisfactory yield of power is dependent on a compromise of several functions. The effects in several aspects of factors affecting combustion are commented on.

Die Elektrische Ausrüstung des Kraftfahrzeuges: Teil 1, Zündung

By Erich Klaiber and Dr. Walter Lippart. Published by M. Krayn, Berlin, Germany. 293 pp.; 213 illustrations. [E-1]

The present volume is Part 1 of a work on automotive electrical equipment and deals solely with ignition. It is a second edition, the original having appeared in 1928, but the changes and improvements that have taken place in the field of ignition have necessitated a complete revision amounting to practically a new treatment of the subject. Greater performance demands and at the same time the urge for economy have brought forth cheaper and more efficient materials, designs and production methods.

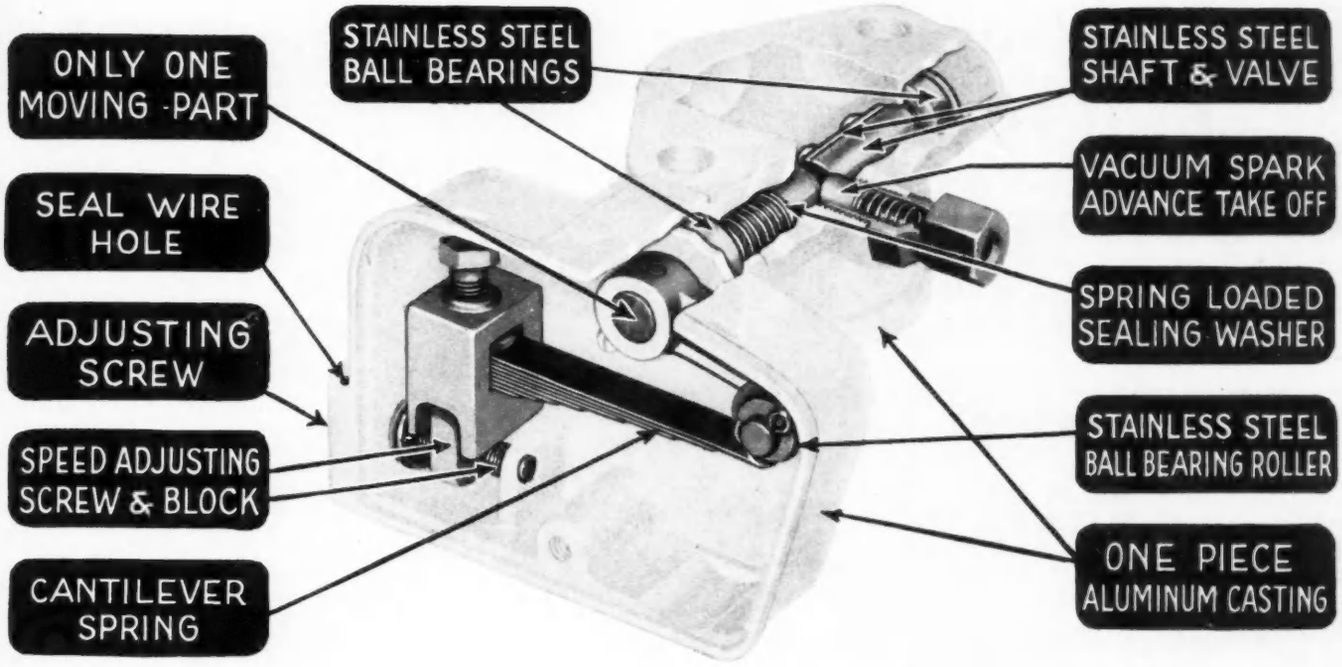
Both theory and practice are covered in the book. In addition to the discussion of fundamentals, the authors describe current ignition equipment and endeavor to weigh without prejudice the relative advantages and disadvantages of the different types. Advice is also given on service and maintenance practice.

The Modern Diesel

Published by Iliffe & Sons, Ltd., London, Third Edition, 1935; 228 pp., illustrated. [E-3]

This book constitutes a review of high speed compression ignition oil engines for road transport, aircraft and marine work, explains with the aid of diagrams the action of these engines, and describes the various fuel injection systems. This edition replaces the volume issued in 1933.

(*Continued on page* 46)



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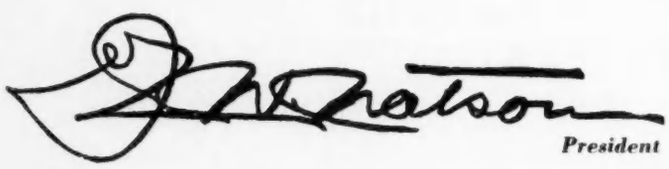
Which shall it be . . . Governors or MORE-MANAGEABLE CARS?

Greater motoring safety is going to be demanded. Personally, I believe that greater safety will result from providing cars with better road-hugging wheel traction and non-pitching and non-swaying bodies than by mere speed government.

In providing this greater and necessary stability, however, we must take no step backward in the matter of comfort. We must retain our present soft springs and their uncanny ability to swallow bumps.

Thoroughly stable, fully manageable cars will do more, I believe, than all else combined to lessen the "toll of the road".

This matter merits the immediate attention of all motor car builders because the weak alternative, speed governors, would take most of the thrill out of buying new models.



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NOTES AND REVIEWS

Continued

Diesel Engines

By J. W. Anderson. Published by McGraw-Hill Book Company, Inc., New York and London, 1935; 491 pp., illustrated. [E-3]

To show the position of the Diesel engine in the United States today, is the aim of the volume as set forth by its author. It describes the various types of engines and classifies them and discusses theory, design, the reasons why the several types and constructions are used, the proper application and installation of the engines with their accessories to suit the service requirements under different operating conditions, operation and maintenance of the equipment, the performance in service and operating costs.

High-Speed Diesel Engines

By Arthur H. Goldingham. Published by E. & F. N. Spon, Ltd., London and Engineers' Book Shop, New York, 1935; 148 pp., illustrated. [E-3]

Mr. Goldingham, author of several previous books on Diesel and oil engines and a contributor to engineering handbooks, presents in this volume a compilation of information on this subject. Considerable space is devoted to the detailed and illustrated descriptions of the various fuel injection systems.

Cylinder Material and Finish with Special Reference to Cylinder Bore Finish from the Users' Point of View as Distinct from that of the Casting Manufacturers'

By K. Brozyna. Published in the *Journal of the Institution of Automobile Engineers,* October, 1935, p. 13. [E-5]

The author concludes that the following are the principal points to be considered in any effort to reduce cylinder bore wear:

(a) Suitable material for cylinders or liners.

(b) Suitable finish and accuracy of bore at all temperatures.

(c) Pistons and rings to be of a suitable design and material hard enough at elevated temperatures; ring grooves and piston ring side faces to be accurate and smooth in conjunction with quick-seating face of ring to prevent gas blowing.

(d) Provision made to quicken warming up of cylinder walls to shorten time of running engine below a certain temperature and to minimize damage caused by cold starting by using means to lubricate top of cylinder.

(e) Use of suitable lubricant and fuel.

(f) At least under severe local conditions use of efficient air cleaners.

MATERIAL

The Protection of Magnesium Alloys Against Corrosion

By H. Sutton and L. F. Le Brocq. Preprint of paper presented before the Institute of Metals, Autumn Meeting, September, 1935, England. [G-1]

Short-time chromate treatments of magnesium-rich alloys of two types in common use were investigated, and the protective value of the treatments compared with the 6-hour chromate treatment and other forms of protective treatment.

A bath has been evolved capable of giving good films in 30 to 45 minutes, and which is suitable for use in steel or aluminum tanks.

The protective value of the 6-hour and short-time chromate treatments, with and without supplementary coatings of varnishes and enamels, was investigated with reference to typical alloys. Intermittent sea-water spray laboratory tests and beach exposure tests were carried out, and the corrosion which occurred observed by loss of weight and change in mechanical properties; and the influence of cleaning treatments employed before chromate treatment was investigated.

Weldability of High-Tensile Steels from Experience in Airplane Construction, with Special Reference to Welding Crack Susceptibility

By J. Müller. Translated from *Luftfahrtforschung,* Vol. XI, No. 4, October 1, 1934; Verlag von R. Oldenbourg, München und Berlin. N.A.C.A. Technical Memorandum No. 779, November, 1935; 25 pp., 12 figs. [G-1]

Proceedings Fifth Mid-Year Meeting American Petroleum Institute Section III—Refining

Published by the American Petroleum Institute, New York City, 1935; 165 pp., illustrated. [G-1]

Papers of particular interest to the automotive industry included in this volume of the Proceedings are: (a) Fuel Specifications for High-Speed Diesel Engines, by G. C. Wilson; (b) "Oiliness" in Motor Oils,

(*Concluded on page* 48)